DIESEL ENGINE DESIGN

BY

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PREFACE TO THE FIRST EDITION

THIS book is based on about twelve years experience of Diesel Engines mainly from the drawing office point of view and is intended to present an account of the main considerations which control the design of these engines

The author ventures to hope that in addition to designer and draughtsmen to whom such a book as this is most naturally addressed there may be other classes of readers—for example Diesel Engine users and technical students—to whom the following pages may be of interest

The text deals mainly with general principles as exemplified by examples of good modern practice and it has not been possible to notice every constructional novelty Apology is perhaps called for on account of the omission of any special treatment of the stepped piston and the opposed piston types of engine These however are the specialities of a comparatively limited number of manufacturers and have been very fully described and illustrated in the technical press

The existence in its fourth edition of Chalkley s well known book on *The Diesel Engine for Land and Marine Purposes* has enabled the present writer to proceed to details with a minimum of preliminary discussion A number of references to other books and papers have been inserted in order to avoid so far as possible overlapping with other sources of information

The author has pleasure in acknowledging his indebtedness to Mr P H Smith (who has at all times placed his unique experience of Diesel Engines at the disposal of the author) for several corrections and suggestions to Mr L Johnson M A (Cantab) for his very careful and patient revision of the proofs to the author s wife for assistance with the manuscript and for compiling the index

H F P P

PREFACE TO THE SECOND EDITION

DURING the past few years there has been a great development in the application of the Diesel Engine particularly to the propulsion of Mercantile Vessels and it is now recognised that for this service there are no necessary limitations of power Further rapid progress is therefore assured in this and no doubt other directions

In order to bring this book into closer touch with later developments it has been necessary to make a large number of minor corrections and some substantial additions

In Chapter VIII some of the more recent forms of cylinders and covers for large engines are described and notes have been added on the influence of combustion chamber shape which have a practical bearing on recent innovations in design

Chapter IX is entirely new and is devoted to heat flow considerations which will necessarily play an important part in the future development of the engine

Chapter XI has been extended by sections dealing with the principles of mechanical injection

Many further references to literature have been added

The writer s acknowledgments and thanks are due to the Editor of *The Motor ship* (London) for kind permission to reproduce Figs 118 122 123 124 125 (together with much of the accompanying text) taken from articles contributed by the author to this journal also to the Hon Sec of the Diesel Engine Users Association for permission to reproduce Figs 110 111 112 119 120 121 126 127 (together with much of the accompanying text) taken from a Paper on Marine Diesel Engines which the author had the honour of reading before the Association in June 1922 The bulk of the material for Chapter IX has been taken from these two sources rearranged and amplified for the purpose of a connected discussion

Finally the author s hearty thanks are also tendered to all those who have kindly helped by pointing out inaccuracies or by offering suggestions

H F P P

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DIESEL ENGINE DESIGN

CHAPTER I

FIRST PRINCIPLES

The Diesel Principle —The characteristic feature of the Diesel Engine is the injection of oil fuel into air which has been previously compressed by the rising of a piston to a pressure corresponding to a temperature sufficiently high to ensure immediate ignition of the fuel

In the course of the pioneer experiments by which the commercial practicability of this engine was demonstrated it was found advantageous to effect the injection of the fuel by a blast of air and this feature was retained in all Diesel Engines until the lapse of the original patents

At the present date there exists a class of high compression oil engines operating on the Diesel principle in which the injection of oil is effected by mechanical means without the assistance of an air blast These engines have been variously termed Solid Injection Engines Cold Starting Heavy Oil Engines Airless Injection Engines For our purpose the term Airless Injection Diesel Engine will serve to distinguish this class from that of the true Diesel Engine as defined below Special features in connection with the design of Airless Injection Diesel Engines will be considered in Chapter XI Throughout the remainder of the book discussion will be confined to the true Diesel Engine in the sense of our definition The well known Hot Bulb Surface Ignition or Hot Plate engines form a very numerous class by themselves and are sometimes known as Semi Diesel Engines they fall outside the scope of this work

The features which characterise the true Diesel Engine in the correct use of the term are now understood to be the following —

(1) Compression sufficient to produce the temperature requisite for spontaneous combustion of the fuel
(2) Injection of fuel by a blast of compressed air
(3) A maximum cycle pressure (attained during combustion) not greatly exceeding the compression pressure i e absence of pronounced explosive effect

Item 3 is deliberately worded somewhat broadly as the shape of a Diesel indicator card is subject to considerable variation under different conditions of load blast air pressure fuel valve adjustment etc

In the earlier days of Diesel Engine construction the square top indicator card shewing a period of combustion at constant pressure was considered the ideal to aim at It has since been found that a card having a more peaked top is usually associated with better fuel consumptions When tar oil is used as fuel the square top card appears to be almost out of the question

It should further be remembered that the existence of a period of combustion at constant pressure is no guarantee that all the combustion takes place at that pressure This ideal is never realised Combustion probably proceeds slowly well after half stroke even under the most favourable conditions

Compression Pressure —The height to which compression is carried is governed by the following considerations —

(1) The attainment of the requisite temperature
(2) The attainment of a desirable degree of efficiency
(3) Mechanical considerations

Considerations of temperature for ignition fix the lower limit of compression at somewhere in the neighbourhood of 400 lb per sq in The temperature actually attained depends on the initial temperature of the intaken air and the heat lost to the jacket during compression so it is clear that the temperature attained on the first few strokes of the engine will be considerably lower than the value it assumes after the engine has been firing consecutively for some time

As regards efficiency it is well known that increasing the degree of compression beyond certain limits does not very materially increase even the theoretical efficiency

In practice the compression most usually adopted is about 500–550 lb per sq in for four stroke engines For two stroke engines the compression is sometimes in the neighbourhood of

600 lb per sq in owing to the fact that the charge of air delivered by the scavenge pump may itself be at a pressure slightly above atmospheric The compression ratio in the working cylinder itself needs to be sufficient to attain ignition temperature since at starting up the scavenge pressure is nearly atmospheric on account of the small pressure required to pass the charge through the ports or valves in the time available The mechanical considerations which limit the compression are numerous and some are mentioned below

Higher compression involves —

(1) Heavier load per sq in of the piston and necessitates massive construction of all the main parts

(2) More highly compressed air for injection and consequently increased trouble with the air compressor and its valves particularly

(3) Increased wear of cylinder liners due to increased pressure behind the piston rings

Compression Temperature —With a compression of 500 lb per sq in in a fair sized four cycle cylinder working under full load conditions the compression temperature is about 1200 F On starting the engine from a cold state the compression pressure and temperature are considerably lower owing to the cold state of the cylinder walls and the piston crown

In addition to this the injection of cold blast air with the fuel in the proportion of about 1 lb of blast air to 12 lb of suction air still further reduces the temperature apart from the probability that the blast air has momentarily a local cooling effect in the zone of combustion

The middle curve (Fig 1) shews graphically the connection between the compression temperature and compression pressure on the assumptions that —

(1) The initial temperature of the intaken air is 212 F

(2) That the exponent in the equation PV^n =const is 1 35

These assumptions correspond approximately to the conditions obtaining with a heavily loaded engine of fair size—say an 18 cylinder with uncooled piston

The noteworthy point about this curve is the slowing down of the rate of increase of temperature with pressure as the latter increases Expressed mathematically $\frac{dT}{dP}$ diminishes as P increases

The Four Stroke Cycle —The well known four stroke cycle consists briefly of —

(1) The Suction Stroke
(2) The Compression Stroke
(3) The Combustion and Expansion Stroke
(4) The Exhaust Stroke

These are considered in detail below

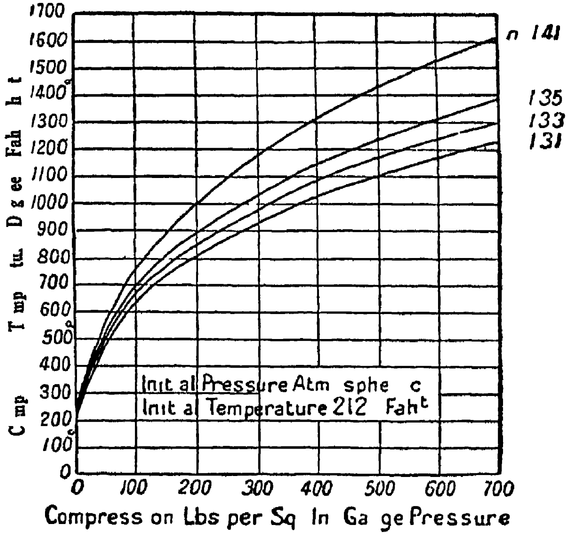


FIG 1

Suction Stroke —If the engine crank is considered to be at its inner dead centre and just about to begin the suction stroke the suction valve is already slightly open In steam engine parlance it has a slight lead At the same time the exhaust valve which has been previously closing on the exhaust stroke has not yet come on its seat The result of this state of affairs is that the rapidly moving exhaust gases create a partial vacuum in the combustion space and induce a flow of air through the suction valve thus tending to scavenge out exhaust gases which would otherwise remain in the cylinder

As the piston descends its velocity increases and reaches a maximum in the neighbourhood of half stroke At the same time the suction valve is being lifted further off its seat and

attains its maximum opening also in the neighbourhood of half stroke The lower half of the suction stroke is accompanied by a more or less gradual closing of the suction valve which however is not allowed to come on its seat until the crank has passed the lower dead centre by about 20 At the moment when the crank is passing the lower dead centre the induced air is passing through the restricted opening of the rapidly closing suction valve with considerable velocity and an appreciable duration of time must elapse before the upward movement of the piston can effect a reversal of the direction of flow through the suction valve It will be clear from the above that owing to the effect of inertia more air will be taken into the cylinder in the manner described than by allowing the suction valve to come on its seat exactly at the bottom dead centre The exact point at which the suction valve should close is doubtless capable of approximate calculation but is usually fixed in accordance with current practice or test bed experiments

Compression Stroke —The piston now rises on its up stroke and compresses the air to about 500 lb per sq in the clearance volume necessary for this compression being about 8% of the stroke volume During the compression the temperature rises and a certain amount of heat is lost to the cylinder walls and cylinder cover The final compression temperature is in the neighbourhood of 900 F to 1200 F

Combustion and Expansion Stroke —At the upper dead centre or slightly previous thereto the injection valve opens and fuel oil is driven into the cylinder and starts burning immediately The actual point at which the fuel enters the cylinder is not quite certain as there is inevitably some lag between the opening of the injection valve and the entrance of fuel The point at which the fuel valve starts to open as determined by a method described below varies from about 3 (slow speed engines) to 14 (high speed engines) The method of determining the point of opening of the fuel valve is as follows —

With the engine at rest air at about 100 lb pressure is turned on to the injection valve and then communication with the blast air bottle is cut off to prevent unnecessary waste of air and the possibility of the engine turning under the impulse of the air which is subsequently admitted to the cylinder The indicator cock is now opened and the engine slowly barred

round by hand until the air is heard to enter the cylinder by placing the ear to the indicator cock. The position of the engine when this occurs is the nearest possible approximation to the true point of opening, assuming the operation has been carefully done.

The duration of the fuel valve opening is usually about 48° and in the majority of engines is fixed for all loads. It is evident that at light load the opening is longer than necessary, and in some designs the duration of opening is regulated by the governor in accordance with the load.

The combustion is by no means complete when the fuel valve closes, and usually continues in some measure well past the half stroke of the engine. This is known as "after burning,"

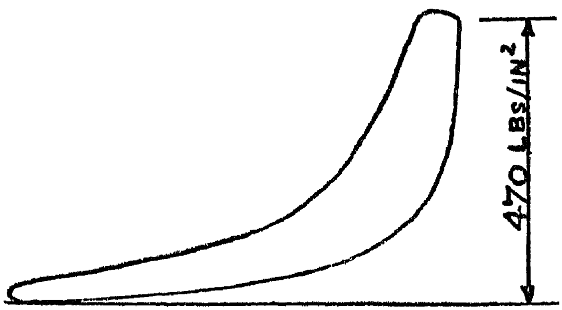


FIG. [illegible]

and takes place with the very best engines in the best state of adjustment. [illegible] after burning is the surest sign of misadjustment, and makes itself apparent by abnormally high terminal pressure at the point at which the exhaust valve opens, and is readily detected on an indicator card by comparison with that taken from an engine in good adjustment. As will be shewn later, the presence of "after burning" is most clearly seen on an Entropy Diagram.

Expansion continues accompanied by loss of heat to the cylinder walls until the exhaust valve opens.

Exhaust Stroke.—The exhaust valve opens about 50° before the bottom dead centre in order that the exhaust gases may effect a rapid escape and reduce the back pressure on the exhaust stroke. The pressure in the cylinder when the exhaust valve starts to open is about 40 lb. per sq. in. with an engine working with a mean indicated pressure of 100 lb. per sq. in. The temperature of the exhaust gases at this point is some

where in the neighbourhood of about 1600 F and the velocity is consequently very high

The pressure falls nearly to atmospheric shortly after the bottom dead centre has been passed and the back pressure during the remainder of the exhaust stroke should not be more than about 1 lb per sq in or less Excessive back pressure may arise from —

(1) Insufficient diameter or lift or late opening of exhaust valve
(2) Exhaust pipe too small in diameter
(3) Obstructions or sharp bends in the exhaust pipe or silencer
(4) Interference by another cylinder exhausting into the same pipe

It is interesting to note that owing to the higher velocity of air at high temperature per unit pressure difference the back pressure is more at light load than at full load

Indicator Cards—Figs 2 and 3 shew typical indicator cards taken with a heavy and a light spring respectively

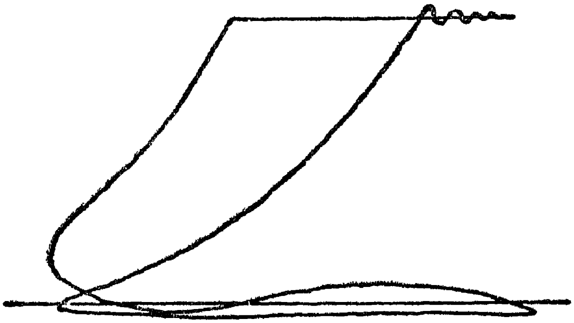


Fig 3

The latter is particularly useful for investigating the processes of suction and exhaust

It is to be observed that in Fig 3 the compression is seen to start at a point which is indistinguishable from the bottom dead centre thus indicating a volumetric efficiency of practically 100% This is to be regarded as a normal state of affairs obtainable with both high speed and low speed engines The volumetric efficiencies of internal combustion engines are frequently quoted at figures varying between about 95% for slow speed engines to 80% for high speed engines The former

figure is reasonable but the latter can only be due either to imperfect design (or adjustment) of the engine or to erroneous indicator cards The use of too weak a spring in the indicator may lead to a diagram shewing not more than 60% volumetric efficiency owing to the inertia of the indicator piston etc Consequently fairly stiff springs are to be preferred

Valve Setting Diagram—Fig 4 is a typical valve setting diagram for a four stroke engine and shews the points relative to the dead centres at which the various valves open and close

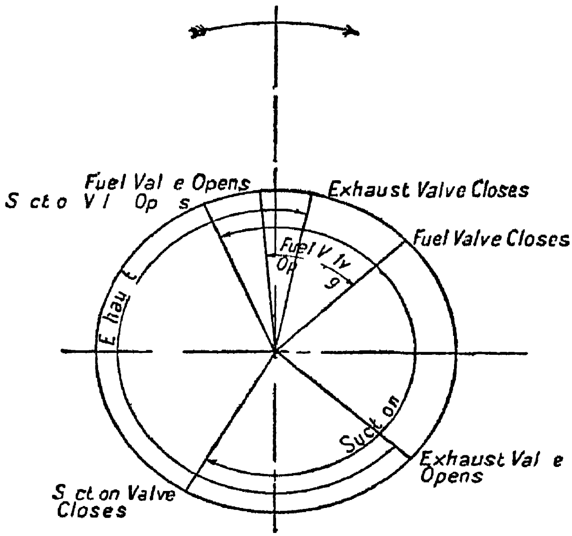


Fig 4

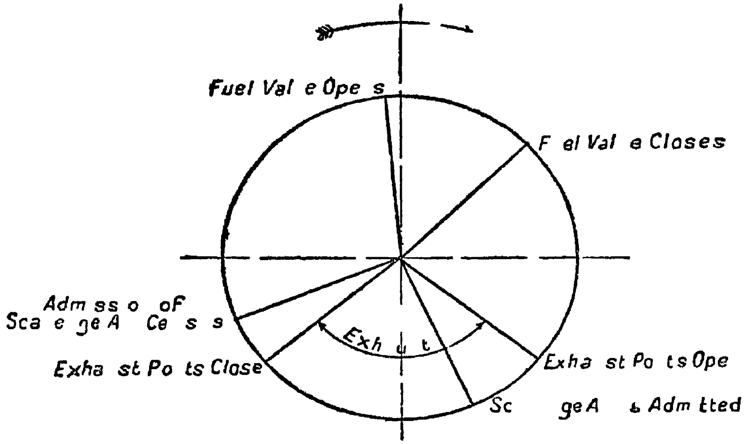
The Two Stroke Cycle—As its name implies the two stroke cycle is completed in one revolution of the engine The revolution may roughly be divided into three nearly equal parts —

(1) Combustion and Expansion
(2) Exhaust and Scavenge
(3) Compression

The exhaust and scavenge take place when the piston is near the bottom dead centre and consequently only very small portions of the expansion and compression strokes are lost in spite of the fact that nearly 120° of the crank revolution are occupied with exhaust and scavenge This point is clearly seen on reference to Fig 5

Exhaust Period —The exhaust starts when the piston uncovers slots in the cylinder wall The point at which this happens is different in different designs of engine an average being about 15% of the stroke before bottom dead centre The exhaust ports are usually of large area and consequently the pressure falls to atmospheric very rapidly The period required for this process naturally depends on the port area and the piston speed and average figures are about 20 to 30

It is well to dwell carefully on the state of affairs at this point



Valve Setting Diagram For 2 Stroke Engine

FIG 5

During exhaust the cylinder pressure has fallen from about 55 to about 15 lb per sq in absolute and there is no reason to suppose that the remaining exhaust gases have fallen greatly in temperature (Given adiabatic expansion the fall in absolute temperature is less than 20%) The conclusions are therefore —

(1) Something like 50% by weight of the gases have effected their escape

(2) The remaining gases are rarefied compared with atmospheric air

Scavenge Period —The scavenge air is admitted by ports or valves (or both) and the instant at which admission starts is timed to coincide with that at which the cylinder contents

attain appreciably the same pressure as the scavenge air or a trifle less The incoming scavenge air is supposed to sweep the remaining exhaust gas before it and so fill the cylinder with a charge of pure air by the time the piston has covered the exhaust slots on the up stroke Actually certain processes take place which do not enter into the ideal programme Some of these are —

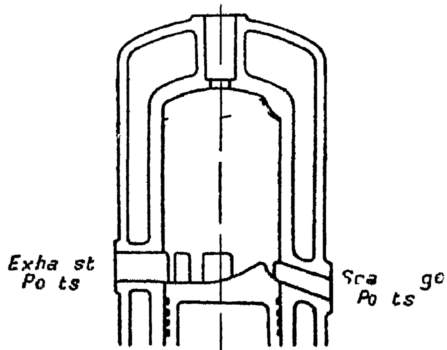
(1) A certain amount of mixing between the incoming scavenge air and the retreating exhaust gases

(2) Short circuiting of scavenge air to the exhaust pipe before all the exhaust gas has been expelled

The effects of both these processes are minimised by providing a large excess of scavenge air The figure adopted for the ratio of scavenger volume to cylinder volume is about 1 4 in modern designs securing good stratification and avoiding undue loss of the fresh charge

There are a number of different systems in use for admitting scavenge air and some of these are discussed below

Simple Port Scavenge —In this system the scavenge air is admitted by means of ports in the cylinder liner opposite a



Diagrammatic Arra gement of
S mple Port Scavenge

FIG 6

row of similar ports for the exhaust (see Fig 6) the piston top being provided with a projection to deflect the scavenge air to the top of the cylinder This system is simple but possesses some disadvantages which are enumerated below

(1) The scavenge air slots have to be made shorter than the exhaust slots in order that the cylinder pressure may fall to

the same value as the scavenge air pressure before the piston begins to uncover the scavenge slots This entails the latter being covered by the piston on its upward stroke before the exhaust ports are covered and consequently the pressure at the beginning of compression can barely exceed the pressure in the exhaust pipe There is also a possibility of exhaust gases working back into the cylinder

(2) The projection on the top of the piston necessitates a specially shaped cylinder cover in order to provide a suitable shape for the combustion space

Engines provided with this system of scavenge are only suited for a relatively low mean indicated pressure of about 80 lb per sq in

Cylinder Cover Valve Scavenge—In this system the scavenge air is admitted by means of one to four valves located in the cylinder cover and avoids some of the disadvantages of the simple port scavenge

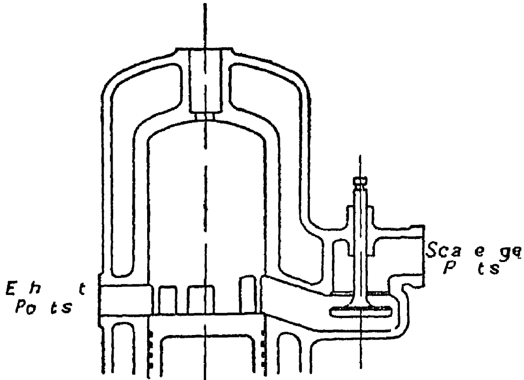
By allowing the scavenge valves to close after the exhaust ports have been covered by the piston the cylinder may become filled with air at scavenge pressure before compression starts and consequently such a cylinder is capable of developing a higher mean effective pressure The greatest drawback to this system is the complication of the cylinder cover and this appears to be rather serious It is not an easy matter to design a cover to accommodate several valves and passages and at the same time secure the necessary conditions for durability under exposure to the strong heat flux which the two stroke cycle involves when high mean pressures are used

Valve Controlled Port Scavenge—This system (usually associated with the name of Messrs Sulzer Bros) appears to combine most of the advantages of both with the disadvantages of neither of the above systems

The air ports or a certain number of the air ports are so situated that they are uncovered before the exhaust ports but are controlled by a valve of the double beat piston or other type in such a way that communication does not exist between the cylinder and the scavenge pipe until the exhaust ports have been uncovered for a sufficient period to allow the cylinder pressure to fall to or below the scavenge pressure On the upward stroke the controlling valve remains open so that the cylinder is in communication with the scavenge pipe until the scavenge slots are covered by the piston (See Fig 7)

Engines controlled on this principle are at present the most successful of the large two stroke Diesel Engines

Amongst small or medium powered two stroke engines the simple port scavenge principle is the favourite



Diagrammatic Arrangement of 2 Stroke Cylinder with Controlled Port Scavenge

Fig 7

Types of Diesel Engines —The existing types of Diesel Engines can be divided into groups in various ways according as they are —

(1) Stationary or Marine
(2) Four Cycle or Two Cycle
(3) Slow Speed or High Speed
(4) Vertical or Horizontal
(5) Single Acting or Double Acting
(6) Air Injection or Mechanical Injection

It suffices here to describe shortly the outstanding features of the commonest types in commercial use

Four Stroke Stationary Engines —The earliest Diesel Engines came in this category and consisted generally of one to three single acting trunk type vertical cylinders having a stroke bore ratio of about 1 4 and running at piston speeds from about 600 to 800 ft per min The cylinders were independent and comprised a cylinder liner lightly pressed into a combined jacket and a frame or column casting Ring lubrication was used for the main bearings Very similar engines are made to day by a large number of makers with 2 3, 4 or 6 cylinders Forced lubrication is frequently used,

and in some cases mechanical injection is employed A number of makers now make use of the enclosed box crank case con struction which is advantageous when forced lubrication is used particularly in conjunction with crossheads Piston speeds generally range from 700 to 1000 ft per minute or over With the higher piston speeds and larger cylinder diameters oil or water cooling of the pistons is resorted to Horizontal engines of the air blast type have been made in considerable numbers abroad but have not found great favour in this country but horizontal engines of the mechanical injection type are now marketed by a number of British makers

Two Stroke Stationary Engines are usually confined to powers of about 700 B H P and upwards They are generally fitted with crossheads Both the open column and closed crank case constructions are in use The earlier engines had cylinder cover valve scavenging but this system is giving way to the port scavenging systems

Four Stroke Marine Engines are now being built by a number of firms foremost amongst which are Burmeister and Wain and their licensees Harland and Wolff For powers up to about 6000 I H P single acting crosshead type engines are used in twin screw ships Special engines having a long stroke and low revolutions are used for single screw vessels on account of propeller efficiency The latest Burmeister and Wain—Harland and Wolff—development consists in the construction of double acting four stroke engines capable of developing 20 000 I H P on two shafts

Four stroke single acting engines with mechanical injection are made by Vickers Ltd for Mercantile Service (crosshead type) and sub marine propulsion (trunk type)

Two Stroke Marine Engines have been built in limited numbers by a number of firms but the leading makers in this field at present are Sulzer and Ansaldo In these engines controlled port scavenge is adopted and the engines are of the normal single acting type The two stroke marine engines made by Cammell Laird and Doxfords are characterised by the use of two opposed pistons per cylinder and in the Doxford engines mechanical injection is used

In the Still engines (two of which at the time of writing have just completed their shop trials) the upper part of each cylinder functions as a two stroke mechanical injection oil engine but steam generated in the cylinder jackets and in

an auxiliary exhaust heated boiler operates on the under side of each piston

Other Types —A number of special types of Diesel Engine have been produced from time to time for various purposes such as locomotive work motor car and aircraft power but these remain in the experimental stage

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CHAPTER II

THERMAL EFFICIENCY

THE overall thermal efficiency of a heat engine is the ratio of the useful work performed to the mechanical equivalent of the heat supplied during a given period of working.

Problem. What is the thermal efficiency of a Diesel Engine which consumes 0.4 lb. of fuel per brake horse power hour?

To solve this problem two things require to be known:—

(1) How much heat 1 lb. of fuel gives out on combustion, i.e. the calorific value of the fuel.

(2) How much mechanical work is equivalent to a given quantity of heat.

The calorific value of different qualities of liquid fuel varies from about 15 500 (Mexican crude) to about 19 300 (Galician crude) British Thermal Units per lb. For calculations and comparison of test results fuel consumptions are usually reduced to their equivalents at a calorific value of 18 000 B.T.U. per lb. One B.T.U. (the amount of heat required to raise the temperature of 1 lb. of water 1° F.) is equivalent to 778 ft. lb. This is Joule's equivalent. Therefore since 1 H.P. hour = 1 980 000 ft. lb. the required thermal efficiency is equal to:—

$$\frac{1\,980\,000}{0.4 \times 18\,000 \times 778} = 0.35$$

From this it is seen that roughly one third of the heat supplied has been converted into useful work. It is within the province of thermodynamics to determine what proportion of the heat loss is theoretically unavoidable, and to what extent the performance of an actual engine approximates to that of an ideal engine working on the same cycle of operations. It is not proposed to give here more than a brief summary of the physical laws relating to the behaviour of air under the in-

fluence of pressure and temperature which form the basis of thermodynamic investigations.

Pressure, Volume and Temperature of Air.—The relation between these three quantities is expressed by the formula:—

$P\ V = 53.2 \times w \times T$—(1) where P = Pressure in lb. per sq. ft. abs.
V = Volume in cubic ft.
w = Weight in lb. of the quantity of air under consideration.
T = Temperature in degrees abs. F.

This relation holds good for any condition of temperature and pressure, and for a specified weight of air, given the values of any two of the quantities represented by capital letters, the third can be calculated.

Example. Find the volume of 1 lb. of air at atmospheric pressure and 60° F. In this case P = 14.7 lb. per sq. in. abs., T = 60 + 461 = 521° abs. F. and w = 1.

$$\text{Hence} \quad V = \frac{53.2 \times 521}{14.7 \times 144} = 13.1 \text{ cub. ft.}$$

Isothermal Expansion and Compression.—If the temperature remains constant during compression or expansion, the process is said to be isothermal, and the value of T in equation (1) becomes a constant quantity. Thence for isothermal processes equation (1) becomes $P\ V = \text{constant}$ (2).

If 1 lb. of air is under consideration the value of the constant is equal to 53.2 times the absolute temperature.

Work Done during Isothermal Compression.—If P_1 and V_1 represent the pressure and volume before compression, and P_2 and V_2 the same quantities after expansion, then:—

$$\text{Work done} = \text{const.}\ \log_e \frac{V_1}{V_2} \qquad (3)$$

the constant being that of equation (2). It should be borne in mind (though the bare fact can only be stated here) that the internal energy of a gas depends on its temperature only, regardless of the pressure. It therefore follows that all the work done in isothermal compression must pass away as heat through the walls of the containing vessel, and for this reason

isothermal processes are not attainable in practice though they may be approximated to by slow compression in cylinders arranged for rapid conduction of heat

Specific Heat at Constant Volume —If 1 lb of air is heated in a confined space i e at constant volume 0 169 B T U are required to raise the temperature by 1 F This then is the specific heat of air at constant volume For many purposes it is near enough to consider the specific heat as constant though actually its value increases slightly as the temperature increases The amount by which the internal energy of 1 lb of air increases as the temperature rises is therefore —

$$0\ 169\ (T_2-T_1) \quad \text{where } (T_2-T_1)=\text{the increase of temperature}$$

Specific Heat at Constant Pressure —In this case on the other hand work is done by expansion if heat is being added and by compression if heat is being discharged consequently the specific heat at constant pressure exceeds that at constant volume by the equivalent of the work done The specific heat at constant pressure is 0 238 B T U per lb per degree Fahrenheit

Adiabatic Expansion and Compression —Expansion or compression unaccompanied by the transfer of heat to or from the air is termed Adiabatic It should be noted that the air may lose heat by doing external work or gain heat by having work done on it by the application of external force during an adiabatic process but this heat comes into existence or passes out of existence within the air itself and does not pass through the walls of the containing vessel

The following relation holds good between the pressure and the volume during an adiabatic process —

$$P\ V = \text{constant} \text{———} (4) \quad \text{where } n = \text{ratio of the two specific heats viz } -1\ 41$$

Owing to the conductivity of the cylinder walls adiabatic compression and expansion are not to be obtained in practice but it is frequently possible to express the actual relation between pressure and volume by means of equation (4) if suitable values are chosen for n A little consideration will shew that for compression with some loss of heat the value of n will be less than 1 41 and for expansion with some loss

c

of heat greater than 1 41 An expansion or compression in which the relation between P and V is expressed by equation (4) is known as a Polytropic Process

Work Done on Polytropic Expansion of Air —The work done is the integral of the pressure with respect to the volume and if expressed in ft lb is given by —

$$W=\frac{P_1V_1-P_2V_2}{n-1}=53\ 2\ \frac{(T_1-T_2)}{n-1} \qquad (5)$$

for 1 lb of air where the suffixes 1 and 2 have their usual significance in denoting the state before and after expansion the work done during a corresponding compression between the same pressures is of course the same

Temperature Change during Polytropic Expansion —By combining equations (1) and (2) the following is obtained —

$$\frac{T_2}{T_1}=\left(\frac{P_2}{P_1}\right)^{1\ -}=\left(\frac{V_1}{V_2}\right)^{\ 1} \qquad (6)$$

With the information supplied by the above six equations it is possible to construct the Indicator Diagram corresponding to an Ideal Diesel Engine in which all defects of combustion and heat losses to the cylinder walls are supposed to be eliminated

An Ideal Diesel Engine —Before proceeding further it is necessary to define what is to be understood by the term ideal engine for the purposes of this investigation In the first place frictional losses and all leakage are eliminated and compression and expansion are supposed to take place adiabatically (n =1 41) The specific heats are supposed to be constant and the same whether for air or a mixture of air or exhaust gases On the other hand the provision of compressed air for the purpose of fuel injection will be recognised and consequently the machine will have a mechanical efficiency less than unity

The compression in the air compressor will be regarded as isothermal and the compressor itself free of all mechanical or other losses

It will also be supposed that all the exhaust gas including that contained in the clearance space is ejected on the exhaust stroke and that every suction stroke fills the cylinder with air at 14 7 lb per sq in abs and 521 abs F The stroke volume is taken for convenience of calculation at 100 cub ft

The Ideal Indicator Card —The indicator card correspond ing to this ideal engine is now readily constructed and is shewn in Fig 8

Point A denotes a volume of air equal to 100 cub ft plus the clearance space which has to be calculated

Line AB represents adiabatic compression from 14 7 lb per sq in to 514 7 lb per sq in abs

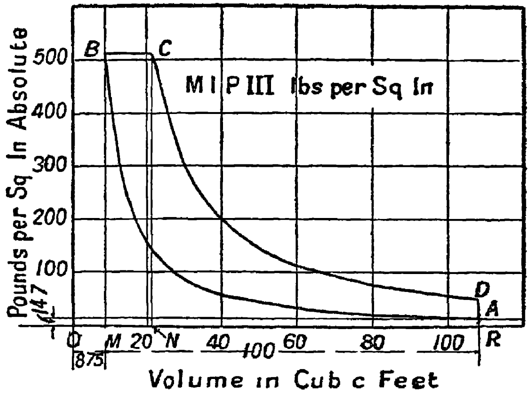


FIG 8

Line BC represents increases of volume at constant pressure due to addition of heat by combustion of fuel

Line CD represents adiabatic expansion of the products of combustion and DA the fall in pressure due to exhaust release It is not necessary to deal with the exhaust and suction strokes as these are supposed to tal e place at atmospheric pressure

Determination of Clearance Volume —

From equation (4)

$$\left(\frac{V_A}{V_B}\right)^{1\ 41} = \frac{P_B}{P_A} = \frac{514\ 7}{14\ 7} = 35$$

$$\frac{V_A}{V_B} = \left(35\right)^{\frac{1}{1\ 41}} = 12\ 45$$

$$\frac{100}{V_B} = 11\ 45 \qquad V_B = 8\ 75 \text{ cub ft}$$

This determines Points B and A

Construction of Line AB —This is effected by calculating the pressure at various points of the stroke in accordance with the following schedule —

1	2	3	4	5	6	7
Per cent of stroke	V = cub ft	$\frac{V_1}{V}$	Log $\frac{V_1}{V}$	(4)×1.41	Antilog (5)	P = (6) × 14.7
0	108.75	1.00	0.000	0.000	1.00	14.7
20	88.75	1.23	0.099	0.127	1.34	19.7
40	68.75	1.58	0.199	0.280	1.90	27.9
60	48.75	2.23	0.348	0.490	3.09	45.5
80	28.75	3.78	0.579	0.815	6.53	9[illegible]
90	18.75	5.79	0.763	1.075	11.89	174.5
95	13.75	7.90	0.898	1.265	18.41	270.6
100	8.75	12.45	1.095	1.545	35.08	514.7

This simple calculation is given in detail, as it is typical.

Determination of Point c.—The value of V depends of course on the amount of heat added. The effect of the oil itself in increasing the weight of the working fluid will be ignored.

The blast air, however, will be taken into consideration and assumed to be equivalent to 8 cub. ft. of free air.

The weight of air concerned has now to be calculated.

(1) Suction air.

From equation (1) $w=\frac{P\times V}{53.2\times T}=144\times\frac{14.7\times 108.75}{53.2\times 521}=8.3$ lb.

(2) Blast air.

$$\text{Weight of blast air}=\frac{8.3\times 8}{108.75}=0.61\text{ lb.}$$

$$\text{suction air}+\text{blast air}=8.3+0.61=8.91\text{ lb.}$$

The effect of the heat released by the combustion of the fuel is to increase the temperature of :—

(1) The suction air (8.3 lb.).
(2) The blast air (0.61 lb.).

at a constant pressure of 514.7 lb. per sq. in. abs.

The temperature of the blast air is taken to be 521° F. abs., and that of the adiabatically compressed air is found from equation (6) as follows :—

$$\frac{T_B}{T_A}=\left(\frac{P_B}{P_A}\right)^{1-\frac{1}{\gamma}}=\left(\frac{514.7}{14.7}\right)^{.29}=2.805$$

$$\text{Therefore } T_B=521\times 2.805=1460^\circ\text{ F. abs.}$$

Now let H =heat added ın B T U
Then sınce the pressure remaıns constant

$$H = 238\ [8\ 3\ (T\ -1460) + 61\ (T_c - 521)]$$
$$= 2\ 12\ T_c - 2961$$

from whıch $$T = \frac{H + 2961}{2\ 12} \qquad (7)$$

and fıom equatıon (1)

$$V = \frac{T\ \times 53\ 2 \times 8\ 91}{514\ 7 \times 144} = 0064\ T_c \qquad (8)$$

Now suppose that 0 2 lb of fuel ıs added havıng a calorıfic value of 18 000 B T U per lb the ıesultıng temperatuıe wıll be —

$$\frac{0\ 2 \times 18\ 000 + 2961}{2\ 12} = 3100\ \text{ F abs}$$

and $V = 0064 \times 3100 = 19\ 8$ cub ft

The expansıon lıne CD can now be constıucted ın the same way as the compressıon lıne both beıng adıabatıcs The value of the termınal pressuıe P_d ıs ıequıred foı calculatıng the woık done and ıs found by means of equatıon (4) as follows —

$$\frac{P_d}{P_c} = \left(\frac{V}{V_d}\right)^{1\ 41} = \left(\frac{19\ 8}{108\ 75}\right)^{1\ 41} = 091$$
$$P_d = 091 \times 514\ 7 = 46\ 8\ \text{lb peı sq ın abs}$$

Calculatıon of Work Done —Referrıng to Fıg 8 the woık done ıs cleaıly equal to the aıeas BCNM plus CDRN less BARM Usıng equatıon (5) foı the two latter we have —

Area BCNM $= 144 \times 514\ 7 \times (19\ 8 - 8\ 75) = 819\ 000$ ft lb

$$\frac{\text{CDRN} = 144(514\ 7 \times 19\ 8 - 46\ 8 \times 108\ 75)}{41} = 1\ 800\ 000 \text{ ft lb}$$

By addıtıon 2 619 000

$$\frac{\text{BARM} = 144(514\ 7 \times 8\ 75 - 14\ 7 \times 108\ 75)}{41} = 1\ 020\ 000$$

Work done by dıfference =1 599 000

Sınce 1 H P hour =1 980 000 ft lb
the fuel consumptıon ıs —

$$\frac{0\ 2 \times 1\ 980\ 000}{1\ 599\ 000} = 248 \text{ lb peı I H P hour}$$

From equation (3) the work done in compressing the blast air is given by

$$8\times 14\ 7\times 144\times \log \frac{914\ 7}{14\ 7} \text{ (assuming blast pressure of 900 lb per sq in)}$$

$$=70\ 000 \text{ ft lb}$$

so the nett work done is

$$1\ 599\ 000-70\ 000=1\ 529\ 000 \text{ ft lb}$$

and the consumption of fuel per B H P hour is —

$$\frac{0\ 2\times 1\ 980\ 000}{1\ 529\ 000}= 259 \text{ lb per B H P hour}$$

The mechanical efficiency being $\frac{1\ 529\ 000}{1\ 599\ 000}=95\ 6\%$

The M I P of the diagram is equal to the indicated work divided by the stroke volume

$$\frac{1\ 599\ 000}{100}=15\ 990 \text{ lb per sq ft}$$

$$=111 \text{ lb per sq in}$$

This is about 10 to 30% higher than the M I P usually indicated at full load in an actual commercial engine

The brake thermal efficiency is given by —

$$\frac{1\ 980\ 000}{0\ 259\times 18\ 000\times 778}=0\ 548$$

A splendid figure for the brake thermal efficiency of an actual engine of large size is 0 35 comparing this with the above figure it is seen that the actual engine attains 64% of the efficiency attributed to the ideal The indicated thermal efficiency of the ideal engine is —

$$\frac{1\ 980\ 000}{0\ 247\times 18\ 000\times 778}=0\ 573$$

That of an actual engine at full power is about 0 472 the ratio of actual to ideal being about 82% From the above it will be seen that so far as the thermal actions within the cylinder are concerned an actual Diesel Engine in good order leaves comparatively little room for improvement as long as the accepted cycle is adhered to As a matter of fact slight deviations from the constant pressure cycle are frequently made and improved efficiencies obtained thereby Most high speed Diesels for example are arranged to give an indicator

card shewing a certain amount of explosive effect i e combustion at constant volume causing the pressure at the dead centre to rise to a figure which may be anything up to about 100 lb above the compression pressure This is found to have a beneficial effect on the fuel consumption which is readily explained on theoretical grounds but it is obvious that considerations of strength must limit the extent to which this principle is used

Fuel Consumption at Various Loads —If the foregoing calculations for the fuel consumption of the ideal engine are

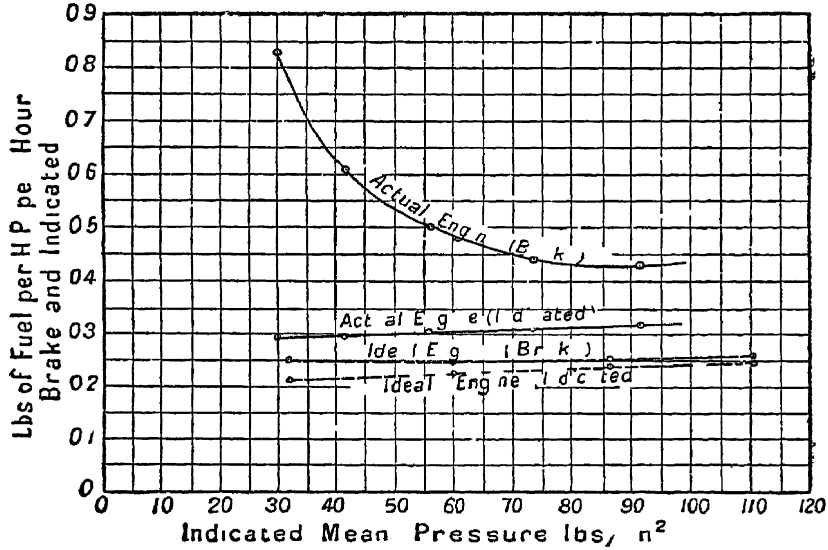


FIG 9

repeated for various values of the quantity of fuel admitted results will be obtained which are shewn graphically in Fig 9 together with test results of an actual engine The fuel consumptions per I H P and B H P hour are plotted on a basis of M I P the actual engine to which the test results refer was of the four stroke type developing 100 B H P per cylinder at full load There are two facts to be noticed —

(1) The fuel consumption per I H P hour increases as the M I P increases

(2) The fuel consumption per B H P hour attains a minimum value at about 90 lb M I P in the case of the actual engine

and about 60 lb in the case of the ideal The reason for (1) will be evident on comparing theoretical diagrams for various values of the M I P and the case is quite comparable to that of a steam engine working with an early cut off

The point of maximum brake efficiency depends upon two conflicting influences viz the indicated efficiency which decreases and the mechanical efficiency which increases as the M I P is augmented

It is usual in this country when considering mechanical efficiency to treat the work done in driving the air compressor as a mechanical loss so that —

$$\text{Mechanical efficiency} = \frac{\text{B H P}}{\text{I H P of main cylinders}}$$

Continental engineers on the other hand sometimes subtract the indicated power of the compressor from that of the main cylinders for the purposes of the above equation

The usual British practice will be adhered to throughout this book

Variation of Mechanical Efficiency with Load —Examination of a large number of Diesel Engine test results reveals the fact that the difference between the I H P and the B H P remains nearly constant as the load is varied This fact enables one to calculate the mechanical efficiency at any load if the full load efficiency is known

Example What is the mechanical efficiency at three quarter half and quarter load if that at full load is 72% ? Let the full load I H P be represented by 100 then the following tabulated figures hold good —

	B H P	I H P	Const Diff	Mech Effcy
Full load	72	100	28	72
Three quarter load	54	82	28	66
Half load	36	64	28	56
Quarter load	18	46	28	39

This method yields sufficiently accurate results for most estimating purposes

Mechanical Losses —The work corresponding to the difference between the I H P and the B H P is approximately accounted for in the two following tables which apply to

medium sized engines of good design and of the four stroke and two stroke types respectively —

Four Stroke Engine—Full Load Mech Efficy 75%	Per cent
Brake work	75 0
Work done on suction and exhaust strokes	3 0
Indicated compressor work	5 8
Compressor friction	1 2
Engine friction opening valves etc	15 0
Work indicated in main cylinders	100 0

Two Stroke Engine—Full Load Mech Efficy 73 5%	Per cent
Brake work	73 5
Indicated compressor work	6 2
Compressor friction	1 4
Indicated scavenger work	3 3
Scavenger friction	1 6
Engine friction opening valves etc	14 0
Work indicated in main cylinders	100 0

Improvements in bearings and guides on the principle of the well known Michell bearing or the use of roller bearings for the main journals and big ends suggest possibilities for reducing engine friction which will possibly materialise in the future The adoption of some form of limit piston ring to prevent excessive pressure on the liners would also help matters in the same direction besides increasing the life of the liners

Influence of Size on Mechanical Efficiency and Fuel Consumption —Piston speed within the range of present practice appears to have very little influence on either mechanical efficiency or fuel consumption This remark does not apply to the abnormally low speeds obtaining for example with a marine engine turning at reduced speed

The cylinder bore is the principal factor in economy always assuming a reasonable ratio of bore to stroke and good design generally

The following table is a rough guide to the mechanical efficiency and fuel consumptions to be expected from cylinders of various sizes working at full load

Four Stroke				Two Stroke			
Cylinder Diam in	Mech Effcy %	Fuel per B H P hr lb	Fuel per I H P hr lb	Cylinder Diam in	Mech Effcy %	Fuel per B H P hr lb	Fuel per I H P hr lb
10	70	46	320	10	68	49	332
15	73	43	315	15	71	46	326
20	75	41	308	20	73	44	322
25	76	40	305	25	74	43	320
30	76	40	305	30	74	43	320

The fuel consumption per B H P at loads other than full load is readily found by first calculating the probable mechanical efficiency as described in the previous article and then allowing for a fall in the consumption per I H P proportional to that shown on Fig 9 for a typical engine for the same M I P

Heat Balance Sheet —An elaborate trial of a Diesel Engine includes the measurement of the quantity of cooling water used the inlet and outlet temperatures of the water and the temperature of the exhaust gases Apart from very slight losses such as radiation etc these data usually enable the heat supplies by the fuel to be accounted for

A typical heat balance is given below —

	Per cent
Accounted for by B H P	34
Rejected to cooling water	28
Rejected to exhaust and otherwise dissipated	38
Total heat supplied	100

A striking feature of this balance is the large amount of heat appearing on the cooling water account which at first sight would appear to indicate very poor utilisation of heat within the cylinder It has been shewn that so far from this being the case an actual engine in good order indicates about 80 % of the work attributable to an ideal engine

The explanation of this lies in the fact that a large proportion of the heat received by the cooling water is given out by the exhaust gases after combustion is complete particularly on their passage through the cylinder cover in the case of a four stroke engine and through the ports in the case of a two cycle Most of the friction work done by the piston and all the compressor work appear on the cooling water account

Efficiency of Combustion —In all actual oil engines there is a considerable amount of after burning i e gradual burning during the expansion stroke In a well tuned Diesel Engine this effect is not sufficiently pronounced to cause smoke even at considerable overload 120 lb per sq in M I P for example Exaggerated after burning is to be avoided as in addition to increasing the fuel consumption it increases the mean temperature of the cycle and of the exhaust stroke particularly and gives rise to accentuation of all the troubles which arise from the effects of high temperature The most prominent of these troubles are enumerated below —

(1) Cracking of piston crowns and cylinder covers
(2) Pitting of exhaust valves
(3) Increased difficulty of lubricating the cylinders resulting in—
  (a) Increased liner wear
  (b) Sticking of piston rings
  (c) Liability to seizure of piston
(4) Increase of temperature of gudgeon pin bearing

The above formidable list is probably not exhaustive but is sufficient to shew the desirability of securing the best possible conditions apart altogether from the question of economy in fuel and lubricating oil consumption The attainment of good combustion assuming a good volumetric efficiency of cylinder and good compression depends more than anything upon small points in connection with the fuel valve which are easily adjusted on the test bed provided the design of the fuel valve is satisfactory

Entropy Diagrams —Entropy is a convenient mathematical concept which it is difficult and perhaps impossible to define in non mathematical terms It is sometimes described as that function of the state of the working fluid which remains constant during an adiabatic process Entropy increases when heat is taken in by the working fluid and decreases when it is rejected If heat is supplied to the working fluid at constant temperature then the increase of entropy is equal to the amount of heat so supplied divided by the absolute temperature If the temperature is variable during the process of heat absorption then the increase of entropy is determined by the integration of the equation —

$$d\phi = \frac{dH}{T}$$ where ϕ = Entropy
H = Heat taken in or given out
T = Absolute temperature

The zero of entropy may be located at any convenient level of temperature except absolute zero. It is convenient for our purposes to consider the entropy to be zero when

$$P = 14.7 \text{ lb. per sq. in. abs.}$$
$$T = 521° \text{ F.}$$

The value of a diagram connecting T and ϕ during the working cycle of an internal combustion engine depends upon the following properties:—

(1) Increasing and diminishing values of ϕ denote heat supplied and heat rejected respectively.
(2) For a complete cycle the diagram is a closed figure the area of which is proportional to the quantity of heat which has been converted into work during the cycle.

The area of the diagram is given by:—

$$\int T\,d\phi = \int \frac{T\,dH}{T} = H_1 - H_2$$ the difference between the heat supplied and that rejected.

Construction of an Entropy Chart for Air (see Fig. 10).—The axis of T is drawn vertically and includes temperatures from 0 to 4000° F. abs. The axis of ϕ is horizontal and is graduated from 0 to 0.35. It will be clear that the axis of T is an adiabatic line of zero entropy and that any selected temperature on this line corresponds to a definite pressure which can be calculated by means of equation (6). Actually it is more convenient to tabulate a series of pressures from 0 to say 700 lb. per sq. in. abs. and tabulate the corresponding temperatures. Points on the axis of T found in this manner are the starting points of constant pressure lines.

Constant Pressure Line $P = 14.7$ lb. per sq. in.—A constant pressure line is a curve connecting T and ϕ when P remains constant. It is usual to consider 1 lb. of air and if the specific heat at constant pressure is assumed to be 0.25.

Then $dH = 0.25\,dT$

$$\text{And } d\phi = \frac{dH}{T} = .25\,\frac{dT}{T}$$

$$\text{Therefore } \phi = .25 \log_e \frac{T_2}{T_1} \text{—} T_1 \text{ being } 521° \text{ F. abs.}$$

Selecting various increasing values of T_2 corresponding values of ϕ are calculated and plotted on the chart a fair curve passing through the points being the required constant pressure line One such line having been constructed lines correspond ing to other pressures are readily drawn since their ordinates

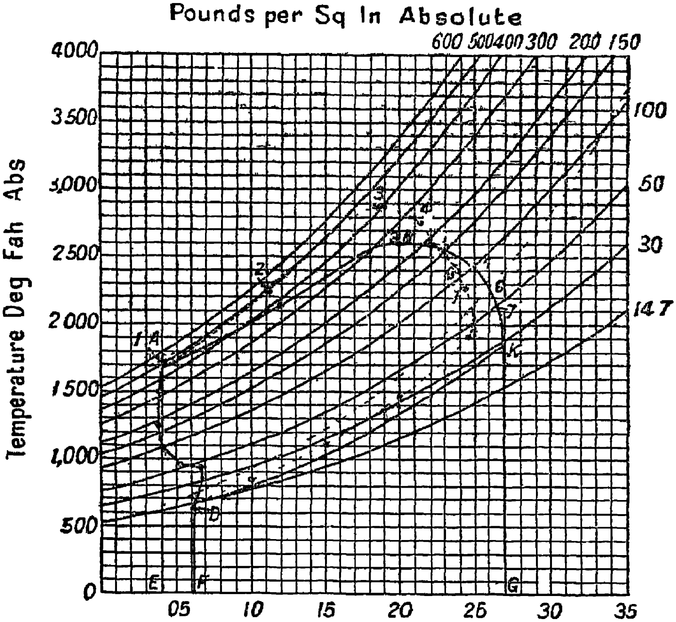


FIG 10

are proportional to the temperatures indicated by their starting points

Constant Volume Lines —These can be constructed in a similar manner using the specific heat at con tant volume instead of that of constant pressure On Fig 10 only one constant volume line is shewn viz that corresponding to

$$P=14\ 7 \text{ lb per sq in}$$
$$T=673 \text{ F abs}$$

as this is required to complete the diagram by representing the rejection of heat at constant volume at the end of the stroke This process is of course a scientific fiction as the pressure in an actual cylinder is reduced to approximately atmospheric pressure by the discharge of exhaust gases and not by cooling

the latter The reason for using the value 0 25 for the specific heat must now be explained Although the specific heat of pure dry air is 0 238 that of the gases present in the cylinder of a Diesel Engine is a variable quantity for the following reasons —

(1) The composition of the working fluid is altered by the addition of the fuel
(2) The specific heat of the exhaust gases increases slightly with increase of temperature

If these variations were rigidly taken into account the construction of the entropy diagram would be a very laborious business and the work is greatly curtailed by adopting certain approximations which will be described In the first place the specific heat is assumed to be constant and equal to the calculated specific heat of the exhaust gases at 60 F The variation in the weight of the working fluid is dealt with by first treating the diagrams as though the weight were constant and then correcting the diagram (entropy) by increasing the entropy and decreasing the absolute temperature of points on the expansion line in the same proportion by which the fuel and blast air increase the weight of the charge

Use of the Entropy Chart —A method of constructing the entropy diagram corresponding to an indicator card will now be described The clearance volume of the engine must first be ascertained and the card accurately calibrated Points on the indicator diagram are then marked corresponding to every 15 or 30 or other convenient division of revolution of the crank past the top dead centre Each of the 12 or more points so marked is given a reference number and the absolute pressure in lb per sq in and also the volume in any arbitrary units (hundredths of an inch on the diagram for instance) corresponding to each point is read off and tabulated The apparent temperature corresponding to each point is then calculated by means of equation (1) on the assumption that the initial temperature just before compression is say 673 F abs

Points on the entropy diagram corresponding to the selected points on the indicator diagram are now found by following the appropriate constant pressure lines until the calculated temperatures are reached thus obtaining the apparent entropy diagram which requires to be corrected in accordance with the preceding article

The entropy diagram shewn in Fig 10 has been constructed in the manner described and the data are given below —

M I P shewn by indicator diagram 82 lb per sq in (see Fig 11)

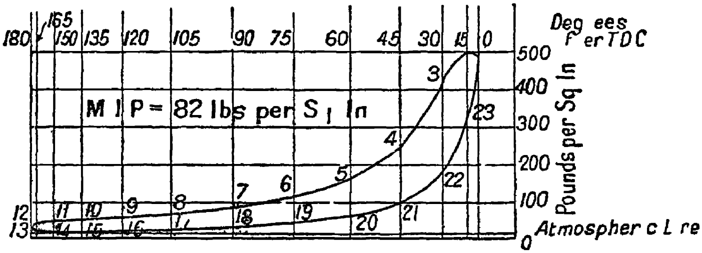


FIG 11

Initial temperature of suction air before compression assumed to be 673 F abs

Absolute temperature (apparent) calculated from PV=kT

where P=Pressure in lb per sq in abs (scaled off diagram)

V=Volume measured in linear inches off diagram

T=Temperature in degrees F abs

Initial conditions are P=14 7 V=6 38 T=673

Whence k= 1395

Figures are given in the table below —

Ref No	Degree after firing centre	V in in	P in lb per sq in	Apparent Temperature
1	0	0 48	490	1690
2	15	0 63	498	2260
3	30	0 95	421	2880
4	45	1 53	251	2760
5	60	2 17	163	2540
6	75	2 94	114	2400
7	90	3 75	84	2260
8	105	4 56	65	2130
13	180	6 38	14 7	673
19	265	2 94	44	930
20	300	2 17	61	950
21	315	1 53	99	1090
22	330	0 95	184	1250
23	345	0 63	331	1500

The apparent entropy diagram was plotted from the above values of P and T and the corrected diagram constructed in accordance with the preceding article is shewn in Fig 10

Area of corrected entropy diagram 9 90 sq in

Temperature scale	1 in =500
Entropy scale	1 in = 05

Therefore work done per lb of suction air is given by —

$$9\ 90 \times 500 \times 0\ 05 = 248 \text{ B T U}$$

1 lb of suction air @ 673 F abs occupies 16 9 cub ft Since clearance volume = 8% of stroke volume therefore corresponding stroke volume = 16 9 − 1 08 = 15 65 cub ft Therefore work done by suction air according to the indicator card @ 82 lb per sq in M I P is given by —

$$\frac{82 \times 144 \times 15\ 65}{778} = 238 \text{ B T U}$$

This figure is about 4% less than that shewn by the entropy chart and suggests that perhaps the assumed temperature of the suction air before compression (viz 673 F abs) is too high which is very probable

Note —The low pressures at the beginning of the compression stroke and the end of the expansion stroke have not been used in the construction of the entropy chart for the following reasons —

(1) Low pressures are very difficult to scale off the indicator card with any degree of accuracy

(2) The low pressures indicated on the card are invariably erroneous unless the greatest care has been exercised in taking the card and that with a suitable indicator in perfect order Many of the reputable makes of indicator though quite suitable for steam engine work give very inaccurate results when applied to internal combustion engines

The following points should be noticed —

(1) The compression line deviates from the true adiabatic in a manner which indicates that heat has been lost to the cylinder walls

(2) The expansion does not become adiabatic until well after half stroke shewing that there is a certain amount of after burning

Area of Entropy Diagram.—If the diagram has been carefully done the area of the diagram in heat units should correspond fairly closely to the work shewn on the indicator diagram. Deviations from equality may be due to :—

(1) Variation in the specific heat at high temperatures.
(2) Incorrect assumption of the initial temperature of the induced air.

Owing to the approximations referred to above there is generally a discrepancy of a small percentage between the amounts of work shewn by the indicator card and the entropy diagram. Also the total area under the upper boundary of the entropy diagram should correspond to the total heat supplied per lb. of suction air less the heat discarded to the water jacket on the expansion stroke. Investigation of the sort described seems to indicate that not more than about 10% to 15% of the total heat supplied by the fuel is lost in this way at 100 lb. M.I.P.

Specific Heat of Exhaust Gases.—The specific heat of a mixture of gases is the sum of the products of the specific heats of the constituents into the proportion by weight in which the constituents are present.

The specific heats (at constant pressure) of the constituent gases present in the exhaust of an oil engine are given below :—

Water vapour	0·480	(Varies considerably with temperature)
Nitrogen	0·247	
Oxygen	0·217	
CO_2	0·210	(Varies considerably with temperature)
CO	0·240	

The average composition of air and the specific heat derived from that of its constituents are given below :—

N_2	75·7%	×0·247 =	·187
O_2	22·7	×0·217 =	·049
H_2O	1·5	×0·480 =	·007
Total	100		·243

This is about 3% higher than the accepted value for pure dry air.

The approximate composition of exhaust gases assuming complete combustion is readily calculated as follows :—

Data. M.I.P.—100 lb. per sq. in.

Fuel pei I H P hour—0 31 lb

Composition of fuel (assumed) $\left\{\begin{array}{l}\text{Carbon } 86\% \\ \text{Hydrogen } 13\% \\ \text{Oxygen } 1\%\end{array}\right\}$ by weight

One H P hour = 1 980 000 ft lb

$$\text{Volume swept by piston per I H P hour} = \frac{1\ 980\ 000}{100 \times 144}$$

$$= 137\ 5 \text{ cub ft}$$

Clearance volume say $8\% = \underline{\ 11\ 0\ }$

Total weight of suction air = 148 5 (assuming perfect scavenge of clearance space)

$$\text{Weight of suction air} = \frac{148\ 5 \times 14\ 7 \times 144}{53\ 2 \times 673} = 8\ 78 \text{ lb @ } 212 \text{ F}$$

Free volume of blast air @ 8% of stroke volume @ 60 F
$= 08 \times 137\ 5 = 11$ cub ft

$$\text{Weight of blast air} = \frac{11 \times 14\ 7 \times 144}{53\ 2 \times 521} = 0\ 83 \text{ lb}$$

Total air = 8 78 + 0 83	= 9 61 lb	= 97%
Weight of fuel	= 0 31	= 3%
Total mixture	9 92	100%

Composition of mixture before combustion is given by —

$$\text{Air} \left\{\begin{array}{ll}N_2 & 76 \times 97 = 73\ 7\% \\ O_2\ (22\ 7 \times 97) + (\ 01 \times 3) & = 22\ 0\% \\ H\ O & 015 \times 97 = \ 1\ 5\%\end{array}\right.$$

$$\text{Fuel} \left\{\begin{array}{ll}C & 86 \times 3 = \ 2\ 5\% \\ H_2 & 13 \times 3 = \ 0\ 4\%\end{array}\right.$$

$$100\ 1\%$$

Combustion of carbon and hydrogen takes place in the proportions given by —

$$C + O_2 = CO_2$$
$$12 + 32 = 44 \text{ by weight}$$

And

$$H_2 + O = H_2O$$
$$2 + 16 = 18 \text{ by weight}$$

The following table shews the composition after combustion and the specific heat (constant pressure) of the mixture —

	Per cent		Specific heat		Product
N_2	73 7	×	247	=	1820
CO_2 $2\ 5 \times 44 - 12$	9 2	×	217	=	0191
O_2 $22 - (32 \times 2\ 5 - 12)$ $-(16 \times 0\ 4 - 2)$	12 1	×	210	=	0254
H_2O $1\ 5 - (18 \times 0\ 4 - 2)$	5 1	×	480	=	0245
	100 1				

Specific heat of exhaust gases = 2510

Repeating the above calculation for different values of the M I P the following figures are obtained —

M I P lb per sq in	0	30	45	75	105	130	155
Specific heat	245	247	248	250	252	254	256

Literature —For thermodynamics of the internal combustion engine consult —

Wimperis H E The Internal Combustion Engine

Judge A W High Speed Internal Combustion Engines

Clerk Sir Dugald The Thermodynamics of Gas Petrol and Oil Engines

In test results see the numerous articles appearing in the technical periodicals and dealing with the recent types of heavy oil engine for stationary and marine purposes more especially *Engineering* and *Motorship*

CHAPTER III

EXHAUST SUCTION AND SCAVENGE

Renewal of the Charge —In the theoretical study of heat engines the charge of working fluid is supposed to remain enclosed in a working cylinder and to undergo a cycle of physical changes due to the introduction of heat from and discharge of heat to external sources by conduction through the walls In actual engines of the internal combustion type one constituent of the charge viz the oxygen takes an active part in the chemical processes which constitute the source of energy In such engines therefore the air charge must be renewed periodically In all existing types of oil engine the charge is renewed as completely as possible for each cycle of thermal changes

The way in which this is done in four stroke and two stroke Diesel Engines has been described in general terms in Chapter I In the present chapter it is proposed to discuss the questions of the discharge of exhaust gases and the introduction of a new air charge from the quantitative point of view apart from the consideration of the mechanical details such as valves cams etc of which these processes involve the use

At the outset it will be necessary to state in a form convenient for application the laws governing the flow of gases through orifices

Flow of Gases through Orifices —Fig 12 is intended to represent a chamber containing gas at a pressure and temperature maintained constant at the values P_1 and T_1 and from which a gaseous stream is issuing through an orifice into the surrounding space which is filled with gas at a constant pressure P_2

In the first instance it will be supposed that P_1 is very much greater than P_2 Then according to the elementary theory of the flow of gases it can be shown that if the gas composing the stream expands adiabatically and all the work done is expended

in increasing the kinetic energy of the stream then the velocity of the latter will increase as expansion proceeds and attain a maximum value given by —

$$V=\sqrt{2g\,J\,K_p\,(T_1-T_2)}=109\ 5\sqrt{(T_1-T\ )} \qquad (1)$$

where —

V =maximum velocity of stream in ft /sec
g =32 2 ft /sec ^{2} (acceleration due to gravity)
J =Joules equivalent (778 ft lb /B T U)
K_p =Specific heat at constant pressure (for air K_p = 238 B T U /lb deg F
T_2 =Temperature attained by the stream after adiabatic expansion from P_1 to P_2 and is given (for air) by —

$$T_2=T_1\left(\frac{P_2}{P_1}\right)^{0\ 285} \qquad (2)$$

In equation (1) the velocity in the chamber is supposed to be negligibly small compared with V For values of P_1 up to

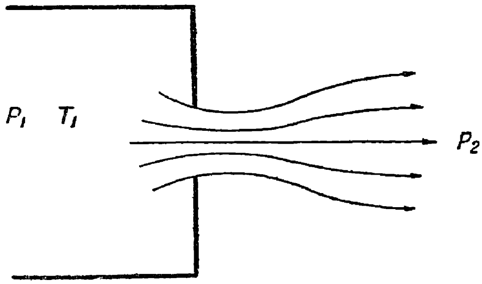


Fig 1

70 × 144 lb per sq ft P_2 being atmospheric equation (1) has been found to agree with experiment within two or three per cent which is ample accuracy for our purpose

It may however be mentioned in passing that experiments at high pressures and particularly with steam indicate that the elementary theory on which equation (1) is based stands in need of correction

It is to be observed that equation (1) is equally valid for any stage of the expansion of the stream so that if any pressure value P be selected lying between P_1 and P_2 then the velocity at this stage will be given by equation (1) with P_a substituted for P_2 If values of the velocity V_A be calculated for various values of P_A and the specific volumes v_A corre

sponding to these values, then the ratios $\frac{v_A}{V_A}$ will be proportional to the areas of the stream at the several stages of expansion. On making such a calculation it will be found that the area of the stream at first contracts and afterwards converges to a final value corresponding to the maximum velocity. In other words, the stream has a neck or throat, as shown in Fig. 12. The pressure P_c at the throat is known as the critical pressure, and for air is equal to $0.53 \, P_1$.

Furthermore, it is evident that if P_2 is equal to P (a contingency which was ruled out at the beginning of the discussion), then the stream will converge to its throat area and remain parallel instead of diverging.

Supposing again that $P_2 < P$, it is evident that for a given size of orifice the discharge will be a maximum if the throat of the stream occurs at the orifice, and in this case the discharge in lb. per sec. will be given by :—

$$Q = A \, V \, W_c \qquad (3)$$

where Q = discharge in lb. per sec.

A = area of orifice—ft.^{2}

V_c = throat velocity—ft./sec.

W_c = weight in lb. per cub. ft. of the air at the conditions of temperature and pressure obtaining at the throat.

So that the discharge is independent of the back pressure P so long as $P_2 < P_c$.

On the other hand, there is no guarantee that the throat of the stream will coincide with the orifice in every case, so that in general the discharge given by equation (3) has to be multiplied by a discharge coefficient less than unity in order to give the discharge observed by experiment.

For air and exhaust gases the following figures may be used :—

Discharge coefficient for sharp edged orifices or ports—0.65
mushroom valves 0.70

The velocity calculated from equation (1) multiplied by the appropriate coefficient may conveniently be called the apparent velocity referred to the actual area of the orifice.

The Suction Stroke.—By way of application of the preceding formulæ, we may consider the suction stroke of a Diesel Engine. The retreating piston creates a partial vacuum and air passes

ın from the atmospheıe vıa the ınlet valve ın consequence Supposıng ıt ıs desıred to lımıt the pressure dıfference to 1 lb per sq ın then—

$$P_1=14\ 7\times 144 \quad P_2=13\ 7\times 144 \quad T_1=\text{say } 520\ \ \text{F abs}$$

$$\text{Then } T_2=520\left(\frac{13\ 7}{14\ 7}\right)^{0\ 285}=509\ 7^{\circ}\ \text{F abs}$$

$$\text{And } V=109\ 5\sqrt{520-509\ 7}=351 \text{ ft /sec}$$

Usıng a coefficıent of 0 7 for the value the apparent velocıty referred to the valve area ıs

$$351\times 0\ 7=246 \text{ ft /sec}$$

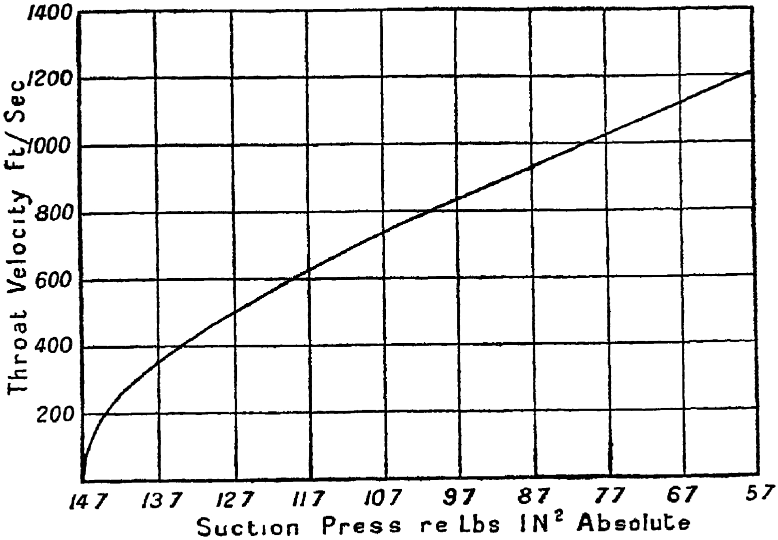


FIG 13

If the pıessuıe dıfference ıs to be constant at 1 lb /ın ² then the value and ıts operatıng cam must be so desıgned that

$$\frac{\text{Instantaneous piston speed (ft /sec)}}{246}=\frac{\text{Instantaneous valve area}}{\text{Piston area}}$$

As ındıcated ın Chapter I ıt ıs better to allow the ınlet valve to open before the ınner dead centre and ın order to obtaın a maxımum charge of aıı the valve should not seat untıl 20 or 30° after the outer dead centıe has been passed

Fıg 13 whıch ıs a curve connectıng calculated velocıty and suctıon pressure ıs the result of repeatıng the above for dıfferent values of P_2 on the assumptıon that

$$P_1=14\ 7\times 144 \text{ and } T_1=520\ \ \text{F abs}$$

Scavenging of Two Stroke Cylinders—Before the scavenge air supply is put into communication with the cylinder whether by valves or ports the exhaust slots should have been sufficiently uncovered for the cylinder pressure to have fallen to practically atmospheric pressure As will be shewn later this takes place with great rapidity owing to the high temperature of the gases On this account the introduction of scavenge air may be assumed to take place against a pressure differing but very slightly from atmospheric This however does not apply to the later stage of the process in those valve scavenged or controlled port scavenge engines in which scavenge air is

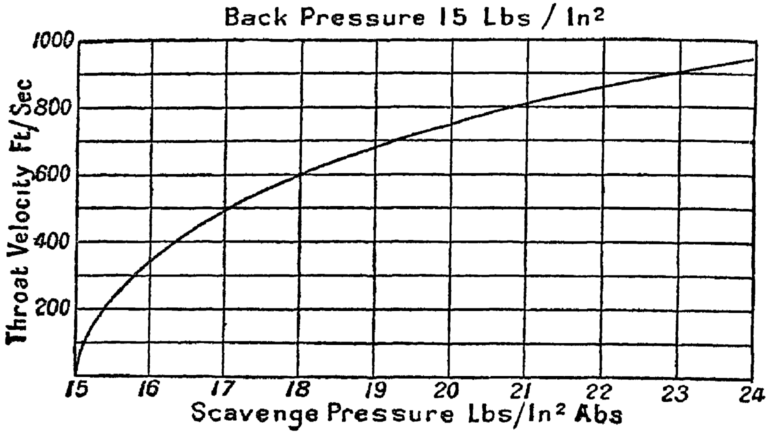


FIG 14

forced into the cylinder after the exhaust ports have been covered During this latter stage the cylinder pressure continually increases until either the scavenge valves (or ports) close and compression begins or until the cylinder contents and the scavenge air supply are in equilibrium

The first stage of the process may be dealt with in a similar manner to that indicated in the preceding article and Fig 14 shews the calculated throat velocity of the scavenge air plotted against the absolute scavenge pressure on the assumption that the back pressure is 15 lb per sq in abs and the temperature of the scavenge air supply is 130 F (591 F abs) this being a good average figure found in practice

Before applying the above to a concrete example some observations will be made on port and valve areas

If a fluid flows for a time t through an orifice of constant

area A with a velocity V then the volume v discharged is evidently given by —

$$v = V\,(A \times t)$$

If on the other hand A is variable then

$$v = V \int A\,dt$$

The quantity $\int A\,dt$ known as the time integral of the area is readily found by plotting A as ordinate against t as abscissæ as in Fig 15 which exhibits the opening area of a valve from the time it lifts ($t=o$) to the time it seats

The time integral of the valve area for the whole interval is the area under the curve For any other interval from say $t=t_1$ to $t=t_2$ the time integral of the area is the area bounded by curve the axis of t and the two ordinates which define

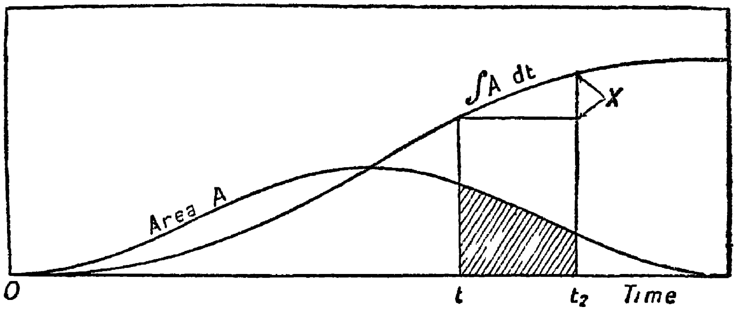


FIG 15

t_1 and t_2 This area is shewn shaded on the diagram for particular values of t_1 and t_2

It is often convenient to take as the unit for t $\frac{1}{100}$th sec or $\frac{1}{1000}$th sec or even 1 degree of revolution of the crank shaft

Also it is very convenient to plot on a t base a curve whose ordinates represent $\int A\,dt$ from $t=o$

This has been done on Fig 15 It will be noticed that between the instants $t=t_1$ and $t=t_2$ the value of $\int A\,dt$ has increased by an amount X The volume discharged between these instants would therefore be (X V)

Example of Scavenge Calculation —Data for two stroke engine —

Bore of cylinder	10
Stroke	15
Revolutions per minute	300

Two scavenge valves 3½ diam maximum lift 1 opening 25 before bottom dead centre and closing 60 after Lift curve harmonic i e the lift plotted on a time base is a sine curve Exhaust ports occupy 60% of the circumference of the cylinder bore and become uncovered by the piston 15% before the end of the stroke The connecting rod is 5 cranks long Fig 16 has been drawn to shew the relation between percentage of

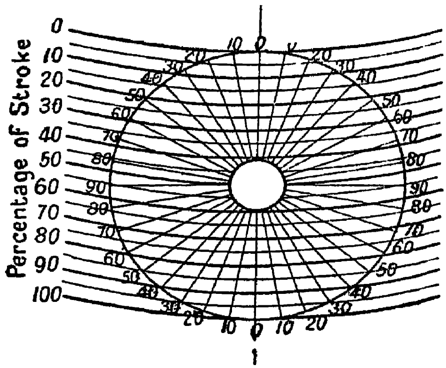


Fig 16

stroke and the number of degrees between the crank position and the bottom dead centre From this diagram it will be seen that the ports are uncovered 50° before B D C and covered again 50 after B D C Compression space 7% of the stroke volume Free air capacity of scavenger 50% in excess of stroke volume of impulse cylinders

$$\text{Stroke volume} = \frac{785 \times 10^2 \times 15}{1728} \qquad = 0\ 68 \text{ ft}^3$$

$$\text{Compression space} = 0\ 07 \times 0\ 68 \qquad = 0\ 05$$

$$\text{Stroke volume} + \text{compression space} \qquad = 0\ 73$$

Volume of scavenge air (at atmospheric pressure and temperature) delivered per cylinder per revolution $= 0\ 68 \times 1\ 5 \qquad = 1\ 02$

Maximum exhaust port area

$$= \frac{0\ 6 \times \pi \times 10 \times 0\ 15 \times 15}{144} \qquad = 0\ 295$$

Maximum scavenge valve area

$$= \frac{2 \times \pi \times 3\ 5 \times 1}{144} \qquad = 0\ 153 \text{ ft}^2$$

Intermediate values of the exhaust port and scavenge valve areas have been plotted on a crank angle base on Fig 17

Number of seconds corresponding to 1 degree of revolution of the crank shaft

$$=\frac{60}{300}\times\frac{1}{360}=0\ 000556 \text{ sec}$$

In Fig 17 values of $\int$A dt for both exhaust and scavenge areas have been plotted in accordance with the preceding article In particular since the scavenge valve lift is harmonic

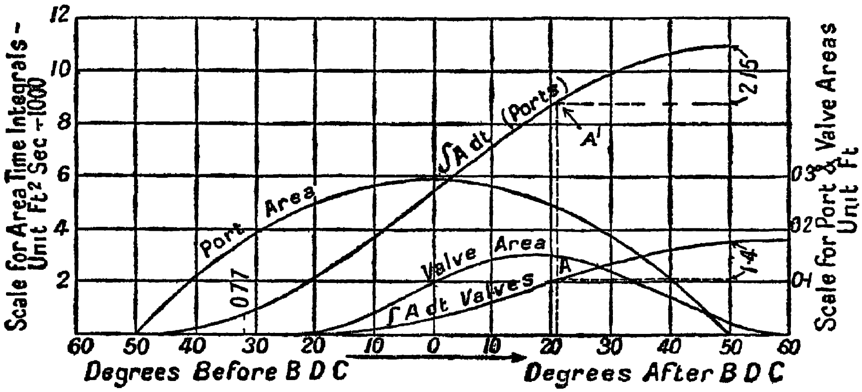


FIG 17

the mean area is one half the maximum and the final value of $\int$A dt is given by —

$$\frac{0\ 153}{2}\times 0\ 000556\times 85\ =10^{3}\times 3\ 61 \text{ ft}^{2} \text{ secs}$$

And since the port area curve is nearly parabolic the maximum value of $\int$A dt for the ports is given by —

$$0\ 295\times\tfrac{2}{3}\times 0\ 000556\times 100\ =10^{3}\times 10\ 94 \text{ ft}^{3} \text{ secs}$$

The intermediate values are readily found by planimetering the areas under the port and valve area curves

Scavenge Air Pressure —A first approximation to the value of the pressure in the scavenger air pipe required by the conditions of the problem is readily obtained on the assumption that the scavenge air is delivered against a constant pressure of 15 lb per sq in abs

Volume discharged	$=1\ 02$ ft ^{3}
$\int$A dt	$=3\ 61\times 10^{3}$

Apparent velocity $= \dfrac{1\ 02 \times 10^{\ 3}}{3\ 61} = 283$ ft /sec

Dividing by a discharge coefficient of 0 7 the calculated velocity is $283 - 0\ 7 = 405$ ft /sec

and the corresponding scavenge air pressure from Fig 14 is —

16 35 lb per in ^{2} abs

This figure cannot however be accepted as final since the effect of discharging the excess of air through the exhaust ports has been ignored

Assume for trial a scavenge pressure of 17 lb per in abs The calculated velocity from Fig 14 is 498 ft /sec and the apparent velocity

$=498 \times 0\ 7 = 349$ ft /sec

The value of $\int A\,dt$ which must be attained to fill the stroke volume and clearance space (0 73 ft ^{3}) is therefore —

$$\frac{0\ 73}{349} = 2\ 09 \times 10^{\ 3}$$

This corresponds to point A on Fig 17 and shews that with the assumed scavenge pressure the cylinder would be filled with scavenge air at about 21 after the bottom dead centre From this point until the point where the exhaust ports are covered three processes are occurring simultaneously viz —

(1) Scavenge air is escaping out of the exhaust ports

(2) Scavenge air is entering the cylinder and raising the pressure of its contents

(3) The piston is rising and reducing the volume of the cylinder contents at the same time tending to raise the pressure

The pressure which exists in the cylinder at the instant at which the piston covers the ports may conveniently be termed the initial charge prèssure and will be denoted by P_i Now if the value of P_i were equal to the scavenge pressure which is its upper limit the amount of air introduced into the cylinder in the interval under consideration would be about equal to —

$$\tfrac{2}{3}\int A\,dt\ (\text{from } 21 \text{ to } 50^\circ \text{ after B D C}) \times 349$$

(for valves)

From Fig 17 $\int A\,dt$ for this inteival is $1\ 4\times10^{3}$ so the volume introduced is

$$\tfrac{2}{3}\times1\ 4\times10^{3}\times349=0\ 326\text{ ft}^{3}$$

Adding to this the volume already introduced viz 0 73 ft ^{3} we obtain 1 056 ft ^{3} which is a trifle in excess of the required quantity

The value assumed for the scavenge pressure viz 17 lb /in ^{2} abs therefore seems to be not far out

The precise value of P_i is a matter of considerable im portance as on it depends the value of the charge weight and consequently the power capacity of the cylinder Its pre determination however appears to be a difficult matter and in practice it is usually adjusted experimentally by advancing or retarding the scavenge cam according as the value of P_i found by light spring indicator diagram is too low or too high

In the former case an adjustment is easily effected by putting a baffling diaphragm in the exhaust pipe

The above calculations however are a sufficient check on the design to ensure the provision of suitable valve or port areas

The volume (@ 15 lb /in ^{2} abs) of scavenge air lost through the exhaust ports is roughly equal to

$$\tfrac{1}{3}\int A\,dt\ (\text{from } 21 \text{ to } 50 \text{ after B D C })\times498\times0\ 65$$
(for ports)

From Fig 17 $\int A\,dt$ for this interval is $2\ 15\times10^{3}$ ft ^{2} sec so the volume required is

$$\tfrac{1}{3}\times2\ 15\times10^{3}\times498\times0\ 65=0\ 23\text{ ft}^{3}$$

So that the air charge is equivalent to $(1\ 02-0\ 23)=0\ 79$ ft ^{3} @ 15 lb per sq in abs

Its actual volume at the point when the piston covers the ports is $0\ 05+(0\ 85\times0\ 68)=0\ 63$ ft ^{3}

and its pressure is therefore about

$$\frac{15\times0\ 79}{0\ 63}=18\ 8\text{ lb /in}^{2}$$

This figure being 1 8 lb /in ^{2} above the assumed scavenge pressure of 17 lb /in ^{2} indicates that the latter figure is in sufficient for the supply of the requisite quantity of air under the conditions specified Further trials on the above lines give a probable value of 18 lb /in ^{2}

Exhaust of Two Stroke Engines —As already mentioned,

it is essential that the scavenge receiver should not be put into communication with the cylinder until the contents of the latter have fallen to a pressure almost equal to that of the scavenge air by the release of the products of combustion through the exhaust slots. This process usually takes a period of time equivalent to 20 to 30 degrees of revolution of the crank shaft. The calculation of this period is much facilitated by the fact that in the interval considered the port area is increasing very approximately in proportion to the time (see Fig. 17). It can easily be shewn that if during an interval from $t=o$ to $t=t_1$ an orifice area increases uniformly with respect to time from O to A_1 and the velocity of efflux also varies uniformly with respect to time from V_0 to V_1, then the discharge is given by :—

$$Q=\frac{A_1T_1}{3}(V_1+\tfrac{1}{2}V_0) \qquad (1)$$

If A represents ft.², t secs. and V ft./secs. of an incompressible fluid, then Q represents cub. ft.

If, as in the case we are about to consider, V represents lb. per sec. per unit of area, then Q represents lb. We proceed to apply equation (1) to the exhaust period of the two stroke engine specified in the previous article.

Exhaust Calculation.—*Data.* Cylinder pressure at the point at which the ports become uncovered 60 lb./in. abs. Temperature at the same point is equal to

$$\frac{\text{initial charge temperature}\times 60}{\text{initial charge pressure}}=\text{say }\frac{670\times 60}{18}$$
$$=2240^\circ\text{ F. abs.}$$

Pressure in exhaust pipe 15 lb./in. abs.

Problem. To find the position of the crank when the cylinder pressure has fallen to 18 lb./in.² abs.

Assume that the charge has the thermal properties of pure air and that the expansion is adiabatic.

At the instant when the ports first become open the volume of the charge is :—

$$0.05+0.85\times 0.68=0.63\text{ ft.}^3$$

and the weight of the charge is :—

$$(60\times 144\times 0.63)\div(53.2\times 2240)=0.0456\text{ lb.}$$

When the cylinder pressure has fallen to 18 lb./in.² abs. the

volume is not certainly known but will not differ much from $0.05+0.92\times0.68=0.675$ ft.³ and its temperature will be —

$$2240\left(\frac{18}{60}\right)^{0.285}=1590°\text{ F. abs.}$$

and the charge weight is reduced to

$$(18\times144\times0.675)-(53.2\times1590)=0.0207\text{ lb.}$$

The weight discharged in the interval is therefore —

$$0.0456-0.0207=0.0249\text{ lb.}$$

The next step is to calculate the rate of discharge per unit of port area.

At the higher pressure of 60 lb./in.² the throat pressure is $0.53\times60=31.8$ lb./in.² abs. and the throat temperature is —

$$2240\left(\frac{31.8}{60.0}\right)^{0.285}=1870°\text{ F. abs.}$$

The calculated throat velocity is therefore —

$$109.5\sqrt{2240-1870}=2110\text{ ft./sec.}$$

and the apparent velocity $=2100\times0.65=1365$ ft./sec.

Now the specific volume at the throat is —

$$\frac{53.2\times1870}{31.8\times144}=21.8\text{ ft.}^3\text{ per lb.}$$

The rate of discharge is therefore —

$$\frac{1365}{21.8}=62.7\text{ lb. per sec. per ft.}^2\text{ of port area.}$$

At the lower pressure of 18 lb./in.² the throat pressure is 15 lb./in.² abs. and the throat temperature is —

$$2240\left(\frac{15}{60}\right)^{0.285}=1510°\text{ F. abs.}$$

The calculated throat velocity is therefore —

$$109.5\sqrt{1590-1510}=978\text{ ft./sec.}$$

and the apparent velocity $=978\times0.65=635$ ft./sec.

Now the specific volume at the throat is —

$$\frac{53.2\times1510}{15\times144}=37.2\text{ ft.}^3\text{ per lb.}$$

and the rate of discharge is therefore —

$$\frac{635}{37.2}=17.1\text{ lb./sec. per ft.}^2\text{ of port area.}$$

We now make the assumption that the rate of discharge per unit area changes its value from the higher to the lower value uniformly with respect to time so that equation (1) can be applied We then have —

$$0\ 0249=\frac{A_1\,t_1}{3}\left(17\ 1+\frac{62\ 7}{2}\right)$$

from which $A_1\,t_1=10\ ^{3}\times 1\ 54$

and $\int A\,dt=\frac{1}{2}\,A_1\,t_1$ (approx)$=10\ ^{3}\times 0\ 77$ ft ^{2} sec

Reference to Fig 17 shews that this value occurs about 18 degrees after the ports begin to open

Alternative Method of Calculation —

Let —

t = time in seconds counting from the instant when the ports begin to be uncovered by the piston

P = pressure of cylinder contents in lb /ft during the period considered so that P varies from 60×144 to 18×144

T = temperature of cylinder contents deg F abs (variable)

w = weight in lb of cylinder contents (variable)

A = port area in ft ^{2} (variable)

v = volume in ft ^{3} of cylinder contents (actually varies during the period considered but treated as constant) $=0\ 65$ ft ^{3}

v = specific volume in ft ^{3}/lb at the throat of the stream issuing from the ports

V = throat velocity in ft /sec

$f(P)=\frac{dP}{dt}-A=$ the rate of pressure drop per unit of port area

then $\frac{dP}{dt}=f(P)\times A$

and $\int A\,dt=\int\frac{dP}{f(P)}$

$f(P)$ has now to be calculated

$$P\,v=53\ 2\,wT \qquad P=\frac{53\ 2}{v}w\,T$$

and $$\frac{dP}{dt}=\frac{53\ 2}{v}\left(T\frac{dw}{dt}+w\frac{dT}{dt}\right)$$

$$=\frac{53\ 2}{v}\left(T\frac{dw}{dt}+w\frac{dT}{dP}\frac{dP}{dt}\right)$$

so that

$$\frac{dP}{dt}=\frac{\frac{53~2}{v}\left(T\frac{dw}{dt}\right)}{\left(1-\frac{53~2}{v}w\frac{dT}{dP}\right)}=\frac{\frac{53~2}{v}T\frac{dw}{dt}}{\left(1-\frac{P}{T}\frac{dT}{dP}\right)}$$

Now $\frac{dw}{dt}=A\frac{V}{v}$

and since $T=T_o\left(\frac{P}{P_o}\right)^{0~285}$ where T and P are the initial values of T and P

$$\frac{dT}{dP}=\frac{0~285~T~P^{(0~285~1)}}{P_o^{0~285}}$$

and $\frac{P}{T}\frac{dT}{dP}=0~285$ (i e constant)

so that $f(P)=\frac{dP}{dt}-A=\frac{\frac{53~2}{v}T\frac{V}{v}}{0~715}$

Values of f(P) are easily calculated for evenly spaced values of P and the integration of $\int\frac{dP}{f(P)}$ can be effected by Simpson s rule

The table on page 50 shews the application of this method to the example of the preceding article

$P=60$ lb /in ² abs $T=2240$ F abs $V=0~65$ ft ³

For convenience P has been expressed in lb /in instead of lb /ft so that a factor of 144 is required Since the interval between successive values of P is 10 5 lb /in ² we have —

$$\int A~dt=\frac{dP}{f(P)}=\frac{10~5\times144\times96~18\times10^{~8}}{3}$$
$$=0~485\times10^{~3}$$

Dividing by the discharge coefficient 0 65 the required value of $\int A~dt$ is —

$$\frac{0~485\times10^{~3}}{0~65}=0~74\ \text{ft}^2\ \text{sec}\ -1000$$

which agrees with the value obtained previously within about 4%

The value of the discharge coefficient (0 65) has been found by comparing the results of calculations similar to the above with the information afforded by light spring indicator cards

1	2	3	4	5	6	7	8	9	10	11	12
P lb /in ² abs	T deg F abs	Throat Press lb /in abs	Throat Temp deg F abs	T mp drop deg F	Throat veloc ty V ft /sec	v	$\frac{V}{v}$	$\frac{53\ 2\ T\ V}{v\ v}$	$\frac{1}{f(P)} = \frac{0\ 715}{[9]}$	Simp so s figure	(10)×(11)
60 0	2240	31 7	1870	370	2110	21 8	96 8	$1\ 78\times10^7$	$4\ 02\times10^{-8}$	1	$4\ 02\times10^{-8}$
49 5	2120	26 2	1770	350	2050	25 1	81 7	$1\ 42\times10^7$	$5\ 04\times10\ 8$	4	$20\ 16\times10\ 8$
39 0	1980	20 6	1655	325	1975	29 8	66 4	$1\ 08\times10$	$6\ 62\times10^{-8}$	2	$13\ 24\times10\ 8$
28 5	1810	15 0	1510	300	1900	37 2	51 1	$0\ 76\times10$	$9\ 44\times10^{-8}$	4	$37\ 76\times10\ 8$
18	1590	15 0	1510	80	980	37 2	26 3	$0\ \ 4\times10$	$21\ 00\times10^{-8}$	1	$21\ 00\times10\ 8$
											$96\ 18\times10^{-8}$

It is also in good agreement with experiments on the flow of gases through sharp edged orifices It is by no means e sy however to obtain light spring indicator diagrams which are at all reliable with an ordinary indicator The fall of pressure is so rapid that the shape of the card is distorted by very little indicator stickiness and oscillations are almost inevitable

With care the latter may be approximately allowed for but cards shewing appreciable indicator stickiness must be rejected These troubles may be largely eliminated by using an optical indicator which is perhaps the only instrument well adapted to this class of investigation

Fig 18 shews a light spring diagram taken from a two stroke Diesel Engine with an ordinary indicator in good order

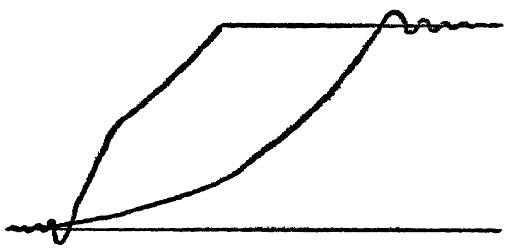


Fig 18

Literature —Funck G Two Stroke Engines with special reference to the Design and Calculation of Ports —*Automobile Engineering* May 1918 *et seq*

Petter H The Escape of Exhaust Gas in Two Stroke Engines —*Engineering* January 4th 1918

Thompson Clarke Research Air Flow through Poppet Valves —Report to U S Nat Advisory Committee for Aeronautics See *Engineering* January 10th 1919

CHAPTER IV

THE PRINCIPLE OF SIMILITUDE

THE properties of similar structures under equivalent conditions have been known for a long time and are implied in most of the empirical formulæ used in machine design At the same time there is a great deal of misconception about the matter in the minds of many to whom a correct appreciation of the principles of similitude would be of considerable value

Definition of Similar Engines—For the purposes of this discussion two or more engines are said to be similar when the linear dimensions of every part of one engine bear a constant ratio to the linear dimensions of the corresponding parts of the other engine or engines Stating the same condition in another way any dimension of any part (the diameter of the gudgeon pin for example) can be expressed as a fraction or proportion of some other dimension (the cylinder bore for example) this fraction or proportion being constant for all similar engines

It follows from the above that two engines to be similar must have the same bore to stroke ratio

In practice there are a considerable number of deviations from strict similarity between different sized engines of the same type built by the same maker a few being noted below —

1 The bore to stroke ratio is subject to variation
2 Thicknesses of metal generally bear a larger fractional ratio to the cylinder bore in small engines than in larger engines
3 Studs bolts and other small gear are made relatively heavier in the smaller sizes to avoid damage by careless handling etc

Equivalent Conditions—Similar engines may be said to be under equivalent conditions when the piston speed is the same and the indicator card identical for both engines

In the past it has been customary to reserve the higher piston speeds for the larger sizes of engines the modern tendency on the other hand is to use the same piston speed for all sizes The former practice appears to have secured uniform durability as measured by the useful life of the engine for all sizes whereas the latter undoubtedly results in the smaller engines being subject to more rapid depreciation than the larger Without appreciable error the indicator cards so far as they influence the stresses in the various component parts of the engine may be taken as identical for all sizes though it is worth mentioning here that it is found advisable in practice to work with a smaller M I P in the case of large cylinders

Properties of Similar Engines —To avoid repetition equivalent conditions will be implied when use is made of the term similar engines unless the contrary is specified Since all parts of similar engines are in proportion their relative size may be expressed by any linear dimension of any part and for this purpose it is very convenient to select the cylinder bore as this is the dimension which assuming constant piston speed determines the power of the cylinder Consider a whole series of similar engines having cylinders of different bores Now the piston load at any point of the stroke will be proportional to the bore^2 since the indicator card is the same for all Now the area of the main bearings will also be proportional to the bore^2 since both linear dimensions contributing to the bearing area are proportional to the bore From this it follows that the bearing pressures are the same for all the engines Without detailing the matter further it suffices to state that with similar engines all the bearing pressures and stresses (including inertia stresses) of corresponding parts are identical Dealing in the same way with rubbing speeds of bearings velocities of gases etc it is found that these also are identical From these facts it follows that given a satisfactory engine other engines made from the same designs but to a different scale (within rational limits) will also be workable machines though for practical reasons such a procedure carried out in minutiæ would not be desirable Actually the foremost makers of Diesel Engines have carried out the principle of similarity remarkably closely

Relative Weight of Similar Engines —Weight being proportional to volume (assuming the same materials are always used for corresponding parts) the weights of similar engines are

proportional to the bore^3 Hence in comparing two different constructions of engine the weight per cubic inch of bore^3 is a convenient criterion of the heaviness of construction Figures for actual engines will be given later Again since the power varies as the bore^2 the weight per horse power varies as the bore with similar engines Owing to departures from strict similarity some of which have been mentioned above the weight per horse power does not in practice increase proportionally as the bore is increased The practice of adopting higher piston speeds for the larger size of engine tends in the same direction with the result that the weight per B H P shews comparatively small variation over a large range of powers

Circumstances which act in the reverse direction are the employment of crossheads and guides and slight reduction of M I P in the larger sizes

In comparing the weight per B H P of Land Diesel Engines the number of cylinders must be taken into consideration for two reasons —

1 The cam shaft driving gear is usually the same for a six or four cylinder engine as for a single cylinder engine of the same bore and type and consequently bears a larger proportion to the total weight in the case of the engine with the smaller number of cylinders The same applies to most of the accessories such as starting bottles fuel filters etc if these are included in the weight

2 The weight of the fly wheel is generally less for a three cylinder engine than for a single cylinder the bore being the same though it will be shewn later that further reduction of fly wheel weight is not advisable for four and six cylinders

A summary of the properties of theoretically similar engines is given below

Piston speed constant M I P constant

1 Linear dimensions proportional to the bore
2 Revolutions per minute inversely as the bore
3 Rubbing velocities and gas velocities are the same
4 Bearing pressures are the same
5 Stresses are the same (including the inertia stresses)

6 Elastic deflections proportional to the bore
7 Natural frequencies of vibration proportional to the R P M
8 Loads (pressure and inertia) proportional to the bore^2
9 Horse power proportional to the bore^2
10 Weight proportional to the bore^3
11 Weight per B H P proportional to the bore
12 Weight per unit of bore^3 the same

The above considerations justify to a great extent the common practice of comparing different designs by expressing

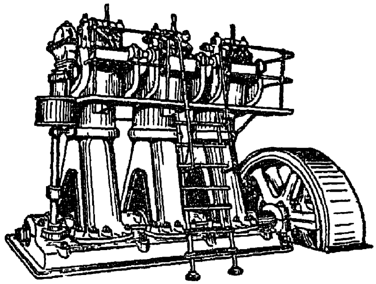


FIG 19 FOUR STROKE LAND ENGINE

the dimensions of corresponding parts as fractions of the bore regardless of whether machines compared are of the same approximate size or not The remainder of this chapter will be devoted to applications

Weight of Diesel Engines —From the purchaser s point of view the weight of a power installation per B H P is a matter of some importance and Fig 20 shews how this quantity varies with different sizes of four stroke land engines of the well known A frame type It will be observed that the weight per B H P increases with the size of the cylinder but not as rapidly as the principle of similitude would lead one to expect This is due in part to the fact that the weights include a quantity of auxiliary gear such as air bottles etc which form a larger percentage of the whole weight in the case of smaller engines The fact of the weight per B H P varying considerably in different sizes renders this figure very unsuitable for estimating purposes The table below shews the weight per cubic inch of bore^3 for the same range of engines and although here also the influence of the weight of the auxiliary gear may

crank case type with trunk pistons and all having three cylinders In this case the fly wheel and auxiliary gear are not included

TABLE II

Bore in in	10 25	12	14	16
Stroke in in	15	18	21	24
Weight in lb (bore$^3 \times 3$)	5 7	5 2	6 6	5 2

From these and other figures it appears that the crank case construction is not intrinsically much lighter than the A frame type of construction and that with the former any economic advantages are traceable to the higher piston speeds which become practicable when the crank case is enclosed

Similar figures are given in Table III for various types of Marine Diesel Engines The figures include the fly wheel and piping attached to the engine but no auxiliary gear

TABLE III

Description	Weight in lb per in of bore $\times$ No of cylind s
Large two stroke Cast iron columns similar to steam engine practice water pumps included	9 5 to 14
Large two stroke Structure of trestle type with through bolts water pumps included	5 2
Large four stroke structure of trestle or crank case type with through bolts water pumps not included	4 5
Large four stroke Structure of stay bolt type	4 0
Trunk engines of crank case type	4 8
Trunk engines of stay bolt type	3 5

For preliminary estimating it is convenient to have an approximate idea of the weights of the piston connecting rod etc of a proposed engine and the following figures are a rough guide to average practice A designer will find it convenient to make notes of similar figures for actual engines with which he has had experience

TABLE IV

Name of Part	Weight per in. of bore lb.
Trunk piston (land engines)	0.18
Connecting rod	0.17
Crank pin	0.036
One web	0.036
Reciprocating weight (land engines)	0.24
Revolving weight	0.18
Piston and piston rod (marine engines)	0.13
Crosshead	0.13
Connecting rod	0.24
Crank pin	0.046
One web	0.046
Reciprocating weight	0.355
Revolving weight	0.236

The above figures are necessarily approximate only as they depend not only on the materials used and the views as to design stresses etc. held by individual designers but also on the bore to stroke ratio and the rules of insurance societies.

Determination of Bore and Stroke.—Nowadays Diesel Engines are generally built up of standardised units in combinations of 2 3 4 6 and 8 cylinders and by this means a large range of sizes may be covered with a minimum of stock patterns jigs etc. The choice of sizes of cylinders and the numbers of different sizes stocked will of course depend on the capacity of the factory and the estimated demand. Marine engines are usually provided with 6 or 8 cylinders in the case of four stroke engines and 4 or 6 cylinders in the case of two stroke engines to facilitate starting and manœuvring and also to keep the size of cylinder and the aggregate weight within reasonable limits. Having decided on the brake power to be developed by a certain standard cylinder the indicated power must be inferred either from previous experience of engines similar to the one proposed by direct estimate of the various losses or with reference to published data of comparable engines. Typical figures for mechanical efficiency have been given on page 26.

A very close approximation of the mechanical losses is obtainable as follows —

1 Air compressor and scavenger losses may be reckoned to be equal to the indicated power of these accessories divided by a mechanical efficiency of 0 8
2 Losses due to friction of piston rings about 5% of the indicated power of the engine
3 Lubricated friction loss is given very approximately by the formula

Horse power lost in lubricated friction

$$=\frac{0\ 3\,[D\,L_1V_1{}^{1\ 5}+n(A+BL_2)V_2{}^{1\ 5}]}{550}$$

Where

D = Diameter of crank shaft in in
L_1 = Aggregate length of journals and crank pins in in
V_1 = Peripheral speed of journals in ft per sec
n = Number of cylinders
A = Area of slipper guide in sq in
B = Bore of cylinder in in
L_2 = Length of piston in in
V_2 = Piston speed in ft per sec

Assuming that the desired indicated power per cylinder is now known the former is related to the cylinder bore by the formula —

$$\text{I H P per cylinder}=\frac{0\ 785\,B^2 P\,V}{2\times 33\ 000}=\frac{B^2\,P\,V}{84\ 000}\text{———}(1)$$

for two cycle engines

And half this amount for four cycle engines both assumed single acting

Where

P = Mean indicated pressure in lb per sq in
V = Piston speed in ft per minute

The piston speed and mean indicated pressure both vary considerably in different cases and existing practice in this respect will be discussed later Table V gives values of the I H P per sq in of bore² for various values of the piston speed and M I P commonly adapted The I H P of a proposed cylinder is found by multiplying the square of the bore by the appropriate coefficient from Table V in the case of a two stroke and by half that figure in the case of a four stroke cylinder

Table V

PISTON SPEEDS—Feet per Minute

M I P lb /in ²	600	650	700	750	800	850	900	950	1000
80	0 572	0 620	0 667	0 715	0 762	0 810	0 857	0 906	0 954
85	0 607	0 658	0 708	0 760	0 810	0 860	0 912	0 962	1 01
90	0 643	0 697	0 750	0 805	0 858	0 912	0 966	1 02	1 07
95	0 680	0 736	0 794	0 850	0 907	0 964	1 02	1 08	1 13
100	0 715	0 775	0 834	0 894	0 953	1 01	1 07	1 13	1 19
105	0 751	0 814	0 876	0 940	1 00	1 06	1 13	1 19	1 25
110	0 786	0 852	0 917	0 983	1 05	1 11	1 18	1 25	1 31

The tabulated figures are values of I H P per in ² of cylinder bore² for two stroke engines For four stroke engines divide the values by 2

Example The I H P of a 30 in four stroke cylinder working with a M I P of 80 lb per sq in and piston speed of 900 ft per minute is —

$$30 \times 0\ 857 - 2 = 386 \text{ I H P}$$

Piston Speeds —For trunk piston land engines of the open type the piston speed usually varies from 600 ft per minute with very small engines of about 8 in bore to 900 ft per minute with large engines of about 23 in bore A linear relation between bore and piston speed between these limits is given by the formula

$$\text{Piston speed} = 460 + 17\ 6\ (\text{bore})$$

which represents good average practice

With high speed forced lubricated trunk engines the practices of the different makers are not so consistent but appear on the average to be based on an increase of about 20% above the figures quoted above for slow speed engines Some makers adopt a uniform piston speed of about 800 ft per minute for all sizes but greater uniformity of reliability would appear to be obtained by a graduated scale of piston speeds as indicated above

Mean Indicated Pressure —Assuming that in every case the engine is or may be required to run continuously at full load then the M I P at rated full load is mainly dependent on the cylinder bore in the case of four stroke engines for the following reasons —

1 Effective cooling of the cylinder walls covers and pistons becomes increasingly difficult as the size of the cylinder is increased on account of the increased length of stream lines through which the heat has to be conducted

2 As the cylinder bore is increased it becomes more difficult to obtain an overload without smoke With a 9 in cylinder it is possible to obtain a M I P of 140 lb per sq in with an invisible exhaust and a rated M I P of 105 lb per sq in is permissible With a 25 in cylinder 120 lb per sq in is about the limit and it is found advisable to restrict the rated M I P to about 87 lb per sq in at nominal full load

Table VI below gives the nominal full load M I P for four stroke engines for continuous running and provides for occasional overloads of short duration amounting to about 20%

TABLE VI

Bore of Cylinder in	8	10	12	14	16	18	20	25	30
M I P at rated full load	110	107	105	102	99	96	94	87	80

Table VI applies more particularly to land engines For large marine engines fitted with cooled pistons slightly higher figures are sometimes used

The M I P admissible in two stroke engines depends on the efficiency of the scavenge process the degree of supercharging and the sufficiency of the arrangements made to minimise heat stresses (see Chapter IX) With efficient scavenging indicated mean pressures of about 125 lb /in ^{2} are obtainable under favourable circumstances with very little if any super charging A service M I P of 90 to 100 lb /in ^{2} is then feasible so far as the provision of an adequate margin for contingencies is concerned but whether an engine can be run continuously at this load with successful results depends on the detail design of cylinders covers and pistons the sufficiency of bearing areas etc

When the scavenge efficiency is poor or indifferent the maximum M I P obtainable is lower and a correspondingly lower value of the service M I P has to be adopted in order to retain a reasonable margin for contingencies

Literature — Diesel Engine Cylinder Dimensions — *Engineering* September 26th 1913

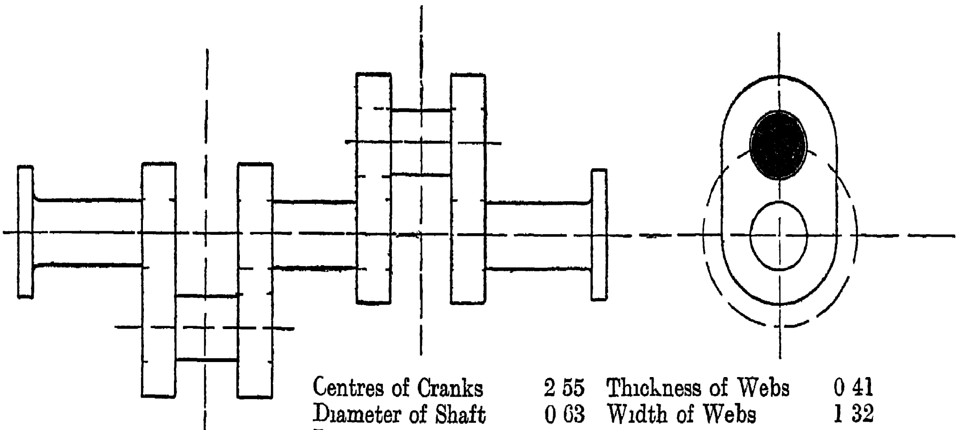
Richardson J The Development of High Power Marine Diesel Engines —Junior I E April 20th 1914 This paper contains a great deal of information regarding the dimensions weights and capacities of the leading types of Diesel Engines

CHAPTEP V

CRANK SHAFTS

Materıal —For Maııne Dıesel Engıne crank shafts the usual materıal ıs open hearth steel havıng a tenacıty of 28 to 32 tons per sq ın and mınımum elongatıon of 25 to 29% ın 2 ın the lower mınımum elongatıon beıng assocıated wıth the hıgher tenacıty

For statıonary engınes ıt ıs moıe usual to employ steel of a tenacıty of 34 tons and upwards specıfyıng a mınımum elongatıon of 25% ın 2 ın



Centres of Cranks	2 55	Thıckness of Webs	0 41
Dıameter of Shaft	0 63	Wıdth of Webs	1 32
Length of Crank pın	0 63		

FIG 21

General Constructıon —Small Dıesel Engıne cıank shafts are usually solıd or semı buılt When the ratıo of stroke to bore exceeds about 1 8 a buılt up shaft ıs sometımes used and typıcal propoıtıons are gıven ın Fıg 21 the unıt beıng the bore of the cylınder

Fıg 22 shews a solıd forged crank shaft foı a two cylınder statıonaıy engıne

For four stroke statıonary engınes the shaft ıs usually of one pıece when the number of cylınders does not exceed four

For six cylinder stationary engines the usual practice is to divide the shaft into two sections arranging the cam shaft drive in the centre Occasionally the one piece arrangement is adopted with the cam shaft drive at one end preferably the fly wheel end This makes the neater arrangement and as no difficulties appear to occur in practice probably the only dis advantage is the expense of replacing the whole shaft in the event of failure The turning of long crank shafts offers no difficulties provided a modern crank shaft lathe is available For four stroke marine engines of six and eight cylinders the usual arrangement is two strictly interchangeable sections of shaft With two stroke marine engines the cylinders are arranged in pairs with a section of shaft to each pair the cranks of which are placed at 180 This arrangement complies

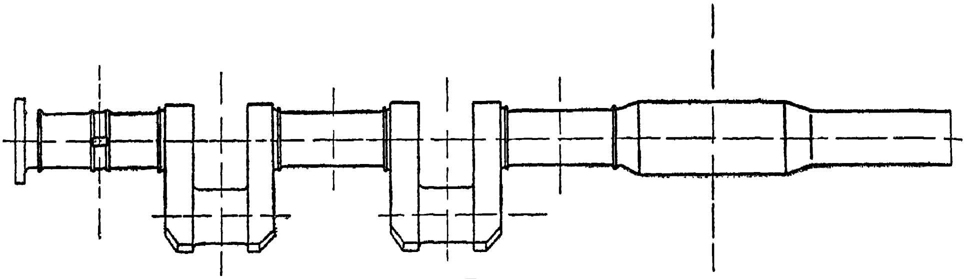


Fig 22

with the requirements of good balance and equal division of impulses and the fact of both cranks of a section being in a plane facilitates both forging and machining To secure inter changeability the number of bolts in each coupling should be either equal to or a multiple of the number of cylinders

Arrangement of Cranks and Order of Firing —

1 Two cylinder four stroke engines

The cranks are commonly arranged on the same centre and the cylinders fire alternately at equal intervals thus sacrificing balance to equal spacing of impulses Arranging the crank at 180 does not very materially affect the degree of uniformity and has the advantage of balancing the primary forces but the primary couples are unbalanced The latter would appear to be the lesser evil

2 Three cylinder four stroke engines

Cranks at 120° and firing periods follow at equal intervals of

240 Primary and secondary forces are balanced but un balanced primary and secondary couples exist The two latter do not appear to be very serious so far as their effects in producing vibrations are concerned

3 Four cylinder four stroke engines

The most common arrangement is to have all the cranks in one plane the inner pair of cranks being on the same centre and the two outer cranks at 180 to them Ignitions follow at intervals of 180 The primary forces and both the primary and secondary couples are balanced but the secondary forces are completely unbalanced By arranging the cranks as for a two cycle engine (*q v*) both primary and secondary forces can be balanced at the expense of unequal spacing of ignitions

4 Six cylinder four stroke engines

In principle each half of the shaft represents the optical image of the other half as seen in a mirror placed at the centre of the engine in a plane at right angles to the centre line of the shaft the cranks in each half being at 120 to each other as for a three cylinder engine The primary and secondary couples generated by each half of the engine mutually cancel each other so that complete balance is obtained so far as primary and secondary forces and couples are concerned

5 Eight cylinder four stroke engines

The same principle of equal but opposite handed shaft halves applies to this case also Each half consists of four cranks the outside pairs of which are at 180 and the planes containing these pairs being at right angles (See Fig 23 which also shews an alternative arrangement)

Each half of the engine is balanced for forces and the two halves balance each other for couples Similar arrangements are possible for higher even numbers of cylinders

6 Four cylinder two stroke engines

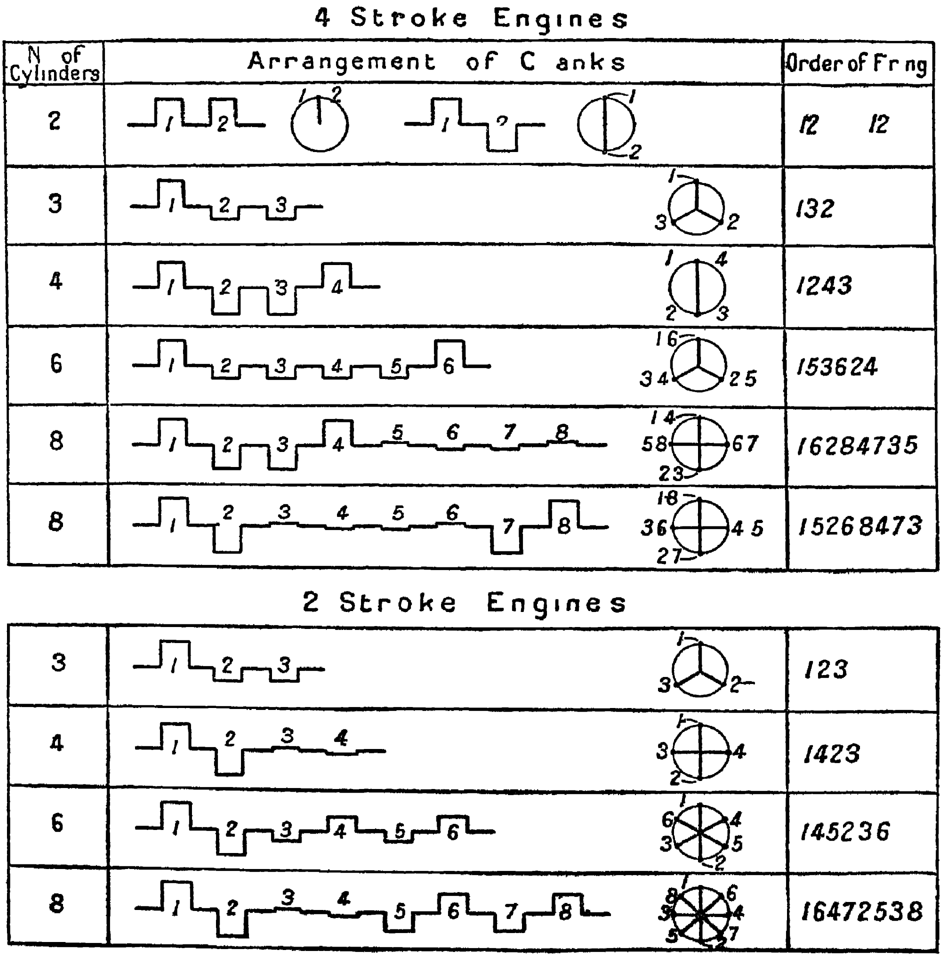
The arrangement for cranks is the same as that described for one half of the shaft for an eight cylinder four stroke engine

7 Six cylinder two stroke engines

The arrangement consists of three pairs of cranks the individuals of each pair being at 180 and the planes contain ing the pairs being at 120° to each other Secondary couples only are out of balance

8 Eight cylinder two stroke engines

Similar to six cylinder engine but planes containing pairs of cranks at angles of 45 Primary and secondary couples out of balance With two stroke engines owing to the fact that no two cranks are on the same centre the order of firing is determined by the angular position of the cranks only With four stroke engines on the other hand having four six or eight cylinders for every firing point there is a choice of two cylinders and the orders of firing generally adopted are based on the principle of placing consecutively firing cylinders as remote as



FIGS 23 24

possible so as to avoid local accumulation of elastic strain due to the reaction at the bearings The usual arrangements and sequences of cranks and also the order of firing are shewn diagrammatically on Figs 23 and 24 for four stroke and two stroke engines respectively

Lubrication —In modern designs of Diesel Engine forced lubrication for the principal bearings is now the rule rather than the exception When crossheads and totally enclosed crank cases isolated from the cylinders are used it is obvious that forced lubrication can be employed with all the convenience and economy which characterises this system as employed in the high speed steam engine

With trunk piston engines certain precautions have to be observed to secure economy

In the first place oil must be prevented from reaching the underside of the hot piston crown This is simply effected by a light diaphragm across the piston a few inches from the crown in fact just clear of the connecting rod top end bearing The diaphragm usually takes the form of a circular cover bolted to an internal flange In some cases this forms a jacket for water or oil cooling of the piston crown With uncooled pistons a sheet steel diaphragm is sometimes used with good effect In any case such a diaphragm by protecting the small end from radiant heat practically eliminates trouble with this bearing

In the second place oil must either be prevented from splashing in undue quantity on to the surface of the liner by means of guards over the cranks or attached to the piston itself or the cylinder mouth or else arrangements must be made to scrape the surplus oil off the liner Holes drilled from the bottom piston ring groove to the inside of the piston are found to be helpful

Finally the crank case must be practically oil tight to prevent loss of oil and the ingress of dirt

So far as economy of lubricating oil is concerned ordinary ring lubrication for the main bearings and the centrifugal banjo arrangement for the big ends leave little to be desired With suitable arrangements for filtering the oil which is drained from the crank pit and using over and over again the net lubricating oil consumption is readily kept below 0 002 lb per B H P hour (trunk piston engines) Similar economies are obtainable with forced lubricated trunk engines when the above precautions are observed Efficient use of non forced

lubrication necessitates certain special features in connection with the crank shaft Rings are turned on the latter at each end of each journal to throw the squeezed out oil into suitable catcher grooves in the bearing brasses whereby it is returned to the oil well instead of being thrown off by the crank webs As these oil throwers have been shewn in some cases to weaken the shaft at its already weakest point they should be designed so as not to interfere with a good radius between the journal and the web Fig 25 shews a section through such an oil thrower

Fig 26 shews a crank fitted with centrifugal banjo lubricator The oil hole leading to the surface of the crank pin is preferably

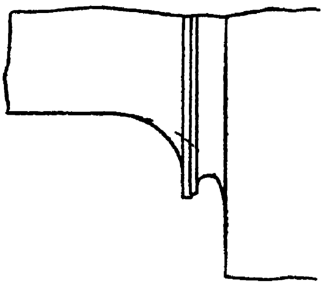


Fig 25

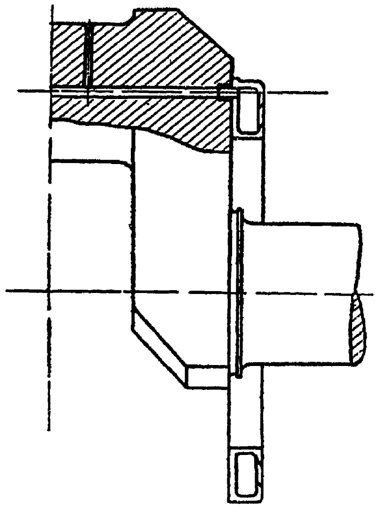


Fig 26

drilled at an angle of about 30 in advance of the dead centre so that the upper connecting rod brass receives a supply of oil just before the ignition stroke

With forced lubrication oil throwers and catchers are not usually fitted but the shaft requires to be drilled to conduct oil from the journals to the crank pins Two systems of drilling are shewn in Figs 27 and 28

Details —(1) **Webs** —Various types of solid forged webs are shewn in Figs 29 30 and 31 The two ends of the straight sided webs are sometimes turned from the journal and crank pin centres respectively Weight can be reduced slightly by turning the two ends at one setting from a centre midway between these two points Triangular shaped segments are

usually turned off the projecting corners of the webs and thi reduced weight facilitates feeling the big ends of the connecting rods and gives more clearance for the indicating gear Balance

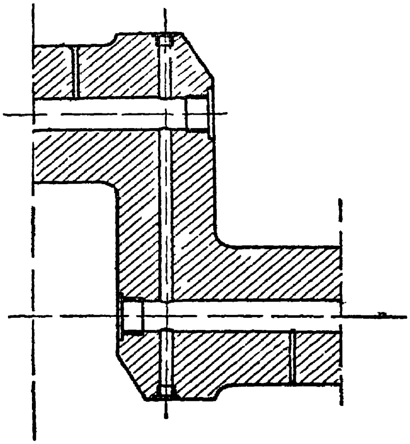


FIG 27

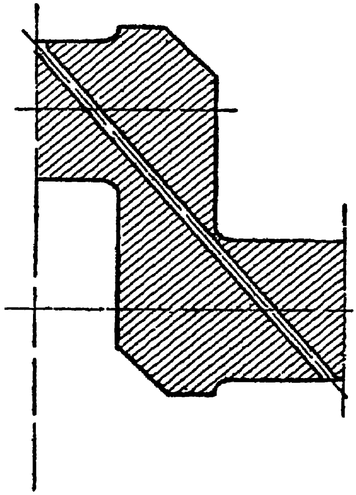


FIG 28

weights are frequently fitted to one and two cylinder engines to balance the revolving weight of the crank pins the big ends and the otherwise unbalanced portion of the crank webs The chief difficulty in designing a balance weight is usually to get a

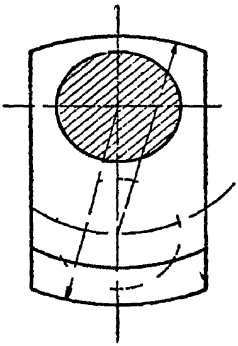


FIG 29

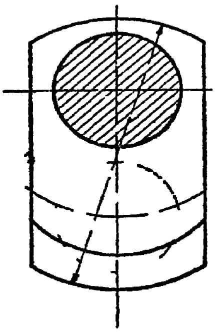


FIG 30

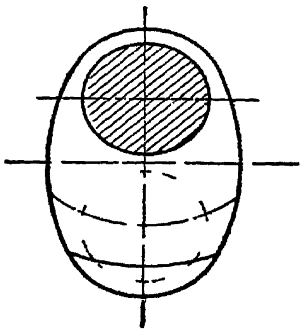


FIG 31

sufficiently heavy weight in the space available The magnitude of the balance weight required is equal to the weight to be balanced (i e weight of crank pin plus about 0 65 of the total weight of the connecting rod and about half the weight of the

webs) multiplied by the radius of the crank and divided by the radius measured from the centre of the shaft to the centre of gravity of the balance weight The problem thus resolves itself into a matter of trial and error Various modes of securing balance weights are illus trated in Figs 32 33 and 34 The bolts or other form of attachment should be suffi ciently strong to carry the centrifugal force of the weight with a low stress

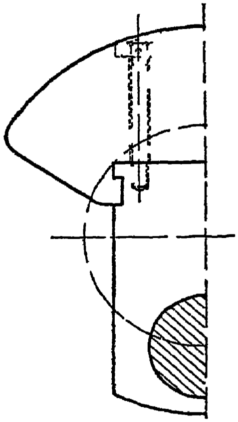


Fig 32

(2) **Couplings** —The couplings connecting the sections of a shaft are made with spigot and faucet joints the spigots being turned off after the bolts have been fitted The bolts belonging to the coupling to which a gear wheel is fitted are usually made of additional length and used to secure the wheel If separate means of securing the wheel are used the interchangeability of the sections of shaft is prejudiced This is of small importance where land engines are concerned With marine engines when it is desired to carry a spare section of shaft the latter

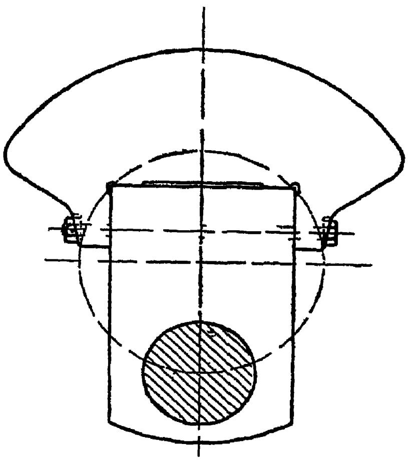


Fig 33

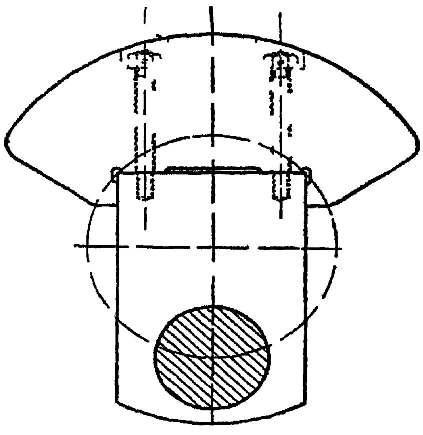


Fig 34

should be provided with any keyways extra bolt holes etc requisite to enable it to replace any section of the shaft in the event of failure with a minimum of fitting

When heavy fly wheels are fitted, as for instance with dynamo

drives an outer bearing is sometimes placed between the fly wheel and the driven shaft The coupling used to connect the projecting end of the crank shaft to the drive may conveniently be of the common cast iron flanged type provided with a shrouding to cover the nuts and bolt heads See Fig 35

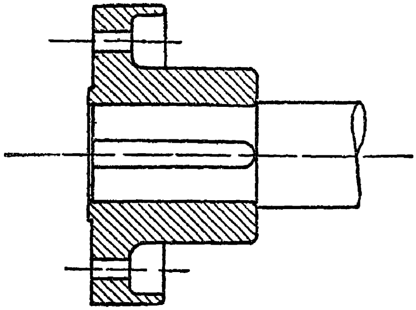


FIG 35

Occasionally this outer coupling is forged with the shaft and if precautions are taken to ensure that the coupling is free of all bending moment it may be proportioned to the twisting moment only This arrangement is a convenient one when the cam shaft drive is at the fly wheel end of the engine as the small diameter of the coupling enables the crank shaft gear wheel to be passed over the coupling in one piece See Fig 36

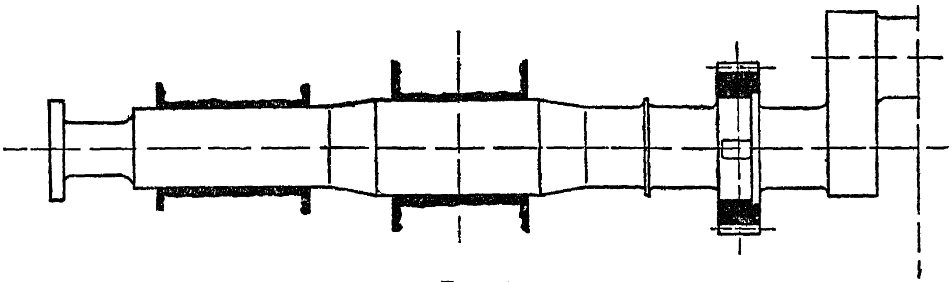


FIG 36

(3) **Air Compressor Cranks** —When the air compressor is supplied by another manufacturer the compressor crank is usually designed by the latter This arrangement is not always quite satisfactory as the apparatus by means of which the compressor is driven at the shop test may be more favour able for satisfactory running than the arrangements which are

made for the reception of the compressor on the engine In any case close co operation should exist between the two manufacturers to produce a satisfactory job between them The main points to be insisted on are rigidity and truth of alignment Figs 37 and 38 illustrate two satisfactory de igns of crank

(4) **Scavenger Cranks** —When the air compressor is driven by an overhung crank extending from the scavenger crank shaft the latter must be made far stiffer than would be required from considerations of strength alone in order to keep the deflection of the overhung crank within small limits In this case it is not unusual to make the scavenger crank the same diameter as the main crank shaft When two scavengers are

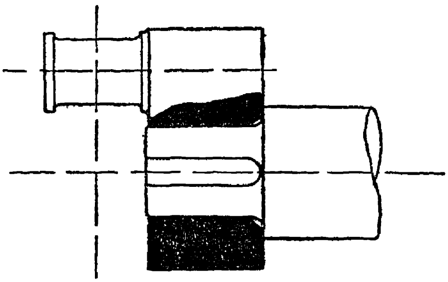


FIG 37

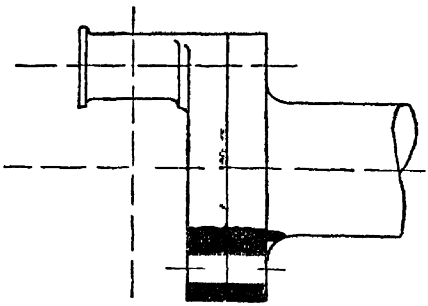


FIG 38

provided the cranks are placed at 90 to equalise the discharge of air

Proportions —In view of the fact that the straining actions causing failure of crank shafts are mainly due to bending moments caused by unequal level of the bearings the most consistent results are obtained when discussing the proportions found in practice by expressing all dimensions in terms of the cylinder bore As regards marine crank shafts the designer has little latitude as minimum dimensions are fixed by the rules of insurance societies To evaluate these rules the distance between centres of cylinders must be determined and this figure is usually about twice the cylinder bore in the case of four stroke engines and about 2 to 2 4 times the bore in the case of two stroke engines The diameter of the shaft usually works out at about 0 62 for four stroke and 0 65 for two stroke engines These approximate figures are merely quoted here for comparison with those relating to land engines

Proportions of Four Stroke Land Engine Crank shafts — Typical proportions for a slow speed engine are given below the unit being the cylinder bore —

Diameter of crank pin and journal	0 53
Length of journal	1 15
Length of crank pin	0 53
Width of web	0 80
Thickness of web	0 28

Proportions of an exceptionally strong shaft are given below —

Diameter of crank pin and journal	0 57
Length of journal	0 88
Length of crank pin	0 70
Width of web	0 92
Thickness of web	0 32

Mr P H Smith in a paper read by him before the Diesel Engine Users Association July 1916 recommends the following proportions applying to 34 ton steel on the understanding that certain precautions are taken to keep the bearings in line

Diameter of crank pin and journal	0 525 to	0 54
Length of journal	0 75	0 80
Length of crank pin	0 524	0 54
Thickness of web	0 32 minimum	

Mr Smith points out that Diesel Engine crank shafts almost invariably fail at the webs and the thickness he proposes for the latter is the greatest the author has found in practice

According to Mr Smith s minimum figure for the shaft diameter and web thickness and taking the width of the web to be 0 8 of the cylinder bore the relative strengths of the journal and the web to resist bending are as

$$\frac{\pi \times 0\ 525^3}{32} \text{ to } \frac{0\ 8 \times 0\ 32^2}{6}$$

$$= 1 \quad 0\ 96$$

so that a shaft based on these proportions will be of nearly equal strength throughout if the effects of radii and changes of shape are neglected Crank shafts for two stroke land engines are generally of slightly larger diameter than those for four stroke engines Different examples give figures varying from 0 55 to 0 59 of the cylinder bore Great difference of opinion exists as to the size of the radii between journals and

crank pins and crank webs A radius of 0 07 of the shaft diameter is good average practice though some designers prefer 0 15 and others are satisfied with 0 04

Calculation of Stresses in the Crank shaft —It is as well to state at the outset that the problem of determining the stresses in a multi crank shaft is rather laborious if done conscientiously and actual designs are more often than not based on experience pure and simple without reference to comparative calcula tions other than of simple proportion Supposing a suitable analysis to have been made for a correctly aligned shaft the whole calculation would require to be revised before the results could be applied with accuracy to the case of a shaft of which the bearings were at different heights owing to the unequal wear of the white metal or flexure of the bedplate This latter consideration is of itself valuable in emphasising the need of massive foundations where land engines are in question and the desirability of providing an extremely rigid framework in marine designs Probably the severest condition with which a Marine Diesel Engine has to contend is the deflection of the hull due to variations of cargo loading etc On this account the engine seatings and the construction of the ship s floor in way of the engines should be strong and stiff The frame work of the engine should also be designed in such a way as to secure a considerable degree of rigidity against sagging or hogging tendencies The component parts of the crank shaft viz journals crank pins and webs are subject to bending and twisting actions which vary periodically as the shaft revolves In the past it has been customary to compute equivalent bending or twisting moments corresponding to the calculated co existing bending and twisting moments and to proportion the shaft accordingly Recent experiments by Guest and others indicate that steel under the influence of combined bending and twisting begins to fail when the shear ing stresses as calculated from the formula quoted below attain a definite value (about 12 000 lb per sq in for very mild steel under alternating stress) which is independent of the relative amounts of bending and twisting

$$\text{Maximum shear stress} = \tfrac{1}{2}\sqrt{4\,f_s^{\,2} + f_n^{\,2}}$$

Where f_n =Normal stress due to bending
f_s =Shear stress due to twisting

The equivalent twisting moment which would give the same shear stress as the maximum shear stress due to the combined action of the actual bending and twisting moments is given by

$$T_E = \sqrt{T^2 + B^2}$$

Where T_E = Equivalent twisting moment
T = Actual twisting moment
B = Bending moment

A good approximation to the twisting moment at any point of the shaft at any degree of revolution is obtained by combining in correct sequence the twisting moment curves corresponding to all cylinders for d of the section under consideration In future the terms forward and aft will be used to denote the compressor and fly wheel end of the engine respectively regardless of whether the engine under consideration is of marine or land type The negative twisting moment due to mechanical friction of the moving parts is almost always neglected That due to the compressor is sometimes allowed for When dealing with the stress in a crank pin it should be borne in mind that the twisting moment due to any cylinder is not transmitted through its own crank pin For example if the cranks are numbered as usual from the compressor end the twisting moment in No 3 crank pin is that due to cylinders Nos 1 and 2 The calculation of bending moments is by no means straightforward and the methods adopted form the distinguishing features of the systems of crank shaft calculation described below

Fixed Journal Method of Crank shaft Calculation —The assumption underlying this method is that each journal is rigidly fixed at its centre and that the section of shaft between two journals may therefore be treated as a beam encastré at its ends The assumption of fixed journals would be true for a row of cylinders all firing at the same time For ordinary conditions however the assumption would only hold good if the bearings had no running clearance and were capable of exerting a bending effect on the shaft by virtue of their rigidity Apart from the fact that the construction of bearings and bearing caps does not suit them for this heavy duty examination of the bearing surface of well worn bearings reveals no trace of such cornering action and justifies the view that the bearings merely fulfil their proper functions of carrying thrust in one direction at a time

Free Journal Method —With this system each crank is supposed to be loaded at the centre of the crank pin and supported freely at the centre of the journals so that the maximum bending moment occurs at the centre of the crank pin This assumption would be approximately true for a single cylinder engine if the weight of the fly wheel and the influence of out board bearings are neglected With this method the twisting moment is of very secondary importance in many cases almost negligible Comparing this system with that described above it will be seen that both involve the construction of twisting moment diagrams though the accuracy of the result is of less importance in the case of the free journal method

In the following articles a four throw crank shaft will be investigated on somewhat different lines with a view to eliminating as many unjustifiable assumptions as possible

Stress Calculation for a Four Throw Crank shaft

Data

Type of engine	four stroke
Number of cylinders	four
Bore of cylinders	10 in
Stroke	15 in
Revolutions per minute	300
Connecting rod	5 cranks long
Maximum pressure at firing dead centre	500 lb per sq in
Diameter of journals and crank pins	5 25 in
Length of journals	8 in
Length of crank pins	5 5 in
Thickness of webs	3 25 in
Width of webs	9 in
Centres of cylinders	8+5 5+6 5=20 in
Weight of piston	170 lb
Weight of connecting rod	190 lb
Weight of crank pin	33 lb
Weight of unbalanced parts of two crank webs	110 lb

The method employed in the following investigation is to calculate the values of the forces acting on the shaft when one crank is at its firing top dead centre The reactions on the bearings will be calculated on the assumption that the centres of the journals remain level and all loads will be treated as concentrated The effect of unequal level of bearings will also

be investigated In computing the forces acting on the crank shaft the dead weight of the latter and also that of the running gear fly wheel etc will be neglected and the effect of the air compressor will not be considered nor will the small exhaust pressure remaining in the cylinder which completes its exhaust stroke at the same instant that the cylinder under consideration begins its firing stroke so that the forces to be dealt with are —

(1) Those due to cylinder pressure
(2) Centrifugal force of revolving parts
(3) Inertia force of reciprocating parts

These will now be calculated —

Weight of revolving part of connecting rod	
0 65 × 190	124 lb
Weight of unbalanced part of crank webs	110 lb
Weight of crank pin	33 lb
Total weight of unbalanced revolving parts	267 lb
Weight of reciprocating part of connecting rod	
0 35 × 190	66 lb
Weight of piston	170 lb
Total weight of reciprocating parts	236 lb

Centrifugal acceleration in the crank circle is —

$$w^2r=\left(\frac{2\pi\ 300}{60}\right)^2\times\frac{7\ 5}{12}=615 \text{ ft per sec}^2$$

Therefore centrifugal effect of revolving parts

$$\frac{267\times 615}{g}=5100 \text{ lb}$$

Inertia effect of reciprocating parts at top dead centre —

$$\frac{236\times 615}{g}\times 1\tfrac{1}{8}=5400 \text{ lb}$$

Inertia effect of reciprocating parts at bottom dead centre —

$$\frac{236\times 615}{g}\times\left(1-\tfrac{1}{8}\right)=3600 \text{ lb}$$

Combined centrifugal and inertia effect at top dead centre

$=5100+5400=10\ 500$ lb upwards

Combined centrifugal and inertia effect at bottom dead centre

$=5100+3600=8700$ lb downwards

Maximum load due to cylinder pressure

$$=0\ 785 \times 10^2 \times 500 = 39\ 000 \text{ lb}$$

Resultant downward effect of pressure centrifugal force and inertia force at firing dead centre

$$=39\ 000 - 10\ 500 = 28\ 500 \text{ lb}$$

Method of Calculating the Reactions at the Bearings — Let A I (Fig 39) represent the centre of the crank shaft A C E G I being the centres of the journals and B D F H those of the crank pins

$R_2\ R_4\ R_6\ R_8$ are the applied forces due to cylinders 1 2 3 4 respectively

$R_1\ R_3\ R_5\ R_7\ R_9$ are the reactions at the bearings unknown in magnitude and direction Under the influence of these forces the centre line of the shaft assumes some deflected shape

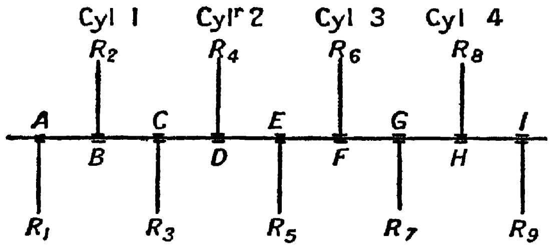


FIG 39

and the deflection at any point above or below the straight line joining A I is equal to the sum of the deflections at the same point which would be produced by each of the forces $R_2\ R_3\ R_4\ R_5\ R_6\ R_7\ R_8$ acting alone supposing the shaft supported freely at A and I At C E and G the sum of these deflections must be zero if the bearings are level (ignoring the effect of running clearance) Let $c_2\ e_2$ and g_2 be the deflections at C E and G due to unit load applied at B (the position of R_2) assuming the shaft supported freely at A and I

$$\left.\begin{array}{c} c_3\ e_3 \text{ and } g_3 \\ c_4\ e_4 \text{ and } g_4 \\ c_5\ e_5 \text{ and } g_5 \\ \text{etc etc} \\ c_8\ e_8 \text{ and } g_8 \end{array}\right\} \begin{array}{l} \text{are the corresponding deflections at C E and G} \\ \text{due to unit load applied at C D E etc} \end{array}$$

The values of $c_2\ e_2$ and g_2 etc are readily found by the usual formulæ for the deflection of beams

Now since the total deflection at C E and G is zero the following equations hold good —

$$R_2 c_2 + R_3 c_3 + R_4 c_4 + R_5 c_5 + R_6 c_6 + R_7 c_7 + R_8 c_8 = 0 \quad (1)$$
$$R_2 e_2 + R_3 e_3 + R_4 e_4 + R_5 e_5 + R_6 e_6 + R_7 e_7 + R_8 e_8 = 0 \quad (2)$$
$$R_2 g_2 + R_3 g_3 + R_4 g_4 + R_5 g_5 + R_6 g_6 + R_7 g_7 + R_8 g_8 = 0 \quad (3)$$

In these three equations the only unknown quantities are R_3 R_5 R_7 which can therefore be determined The remaining unknown reactions R_1 and R_9 are found by equating moments about I and A respectively In solving the above equations downward forces and deflections will be considered positive and upward forces and deflections negative

Determination of c_2 e_2 g_2 etc — In determining these constants it will be assumed that the shaft deflects under load as though it were a cylindrical beam of the same diameter as the crank pins and journals Considering that the webs of a Diesel Engine crank are short the presumption is probably not inaccurate but it would be interesting to see this point investigated as it could readily be by means of models So long as bearings at constant level are assumed the actual value of the deflection is of no importance as the method of calculation depends only on the relative deflection at the different points considered The problem therefore resolves itself into finding the deflected form of a beam freely supported at each end under the influence of a concentrated load placed anywhere between the supports This may be done by treating each end of the beam as a cantilever The deflection of the end of a cantilever carrying a load at the end is given by —

$$\text{Deflection at the end of cantilever in in} = \frac{W\,l^3}{3\,EI}$$

Where W = Load in lb
E = 30 000 000 lb per sq in (for steel)
I = Moment of inertia (transverse) of the section of beam in

For the shaft under consideration —

$$I = \frac{\pi}{64} \times 5\ 25^4 = 37\ 3 \text{ in}^4$$

Fig 40 shews the values of the deflection at various fractional points in the length of the cantilever the deflection at the end being unity These results are applied to the case of a beam as follows —

Let AB be a beam (Fig 41) supported at A and B and

carrying a load W_1 at any point C The reaction at A is equal to $W_1 \frac{CB}{AB}$ Let this be denoted by W_2 At A erect a perpendicular AD equal to some convenient scale to the deflection of the cantilever AC due to the load W_2 at its end Draw the deflected shape of this cantilever (DC) by means of the proportions given

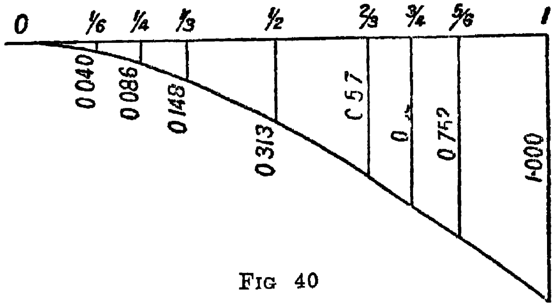


Fig 40

in Fig 40 Proceed similarly with cantilever C B obtaining the deflected shape C F Join D and F then the deflection of the beam at any point X is the vertical intercept x shewn in the figure

Application to the Case in Hand —The constants c_2 e g_2 being the deflections at various points on a beam due to unit load applied at other various points are independent of the system of loads and will therefore be dealt with before special

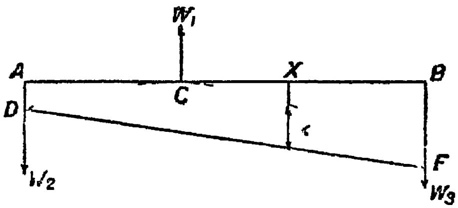


Fig 41

cases of loading are considered In order to obtain manageable figures the deflection will be reckoned in thousandths of an inch and the unit load will be taken as 10 000 lb Deflection of cantilevered portion of shaft in thousandths of an inch is given by —

$$\frac{W\left(\frac{l}{10}\right)^3}{3 \times 30 \times 37\ 3} = \frac{W\left(\frac{l}{10}\right)^3}{3360}$$

Where W = Load in lb

l = Length in in

The diagram Fig 42 shews the process of determining $c_2 e_2 g_2$ etc in particular —

Unit load at B = 10 000 lb AB = 10 BI = 70

$$\text{Reaction at A} = \frac{10\ 000 \times 70}{80} = 8750 \text{ lb}$$

$$\text{I} = \frac{10\ 000 \times 10}{80} = 1250 \text{ lb}$$

$$\text{Deflection of cantilever AB} = \frac{8750 \times 1^3}{3360} = 2\ 6_{\overline{000}}$$

$$\text{IB} = \frac{1250 \times 7^3}{3360} = 127_{\overline{100}}$$

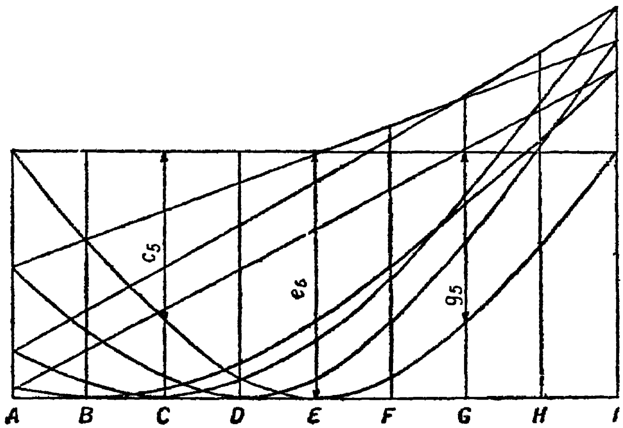


Fig 42

Deflected shapes of cantilevers drawn by plotting ordinates from the proportion given in Fig 40

The constants are found to be —

$$c = 30\ 2 \quad e_2 = 35\ 0 \quad g_2 = 22\ 0$$

Since the deflection at G due to a load at H is the same as that at C due to the same load at B and so on therefore —

$$g_8 = 30\ 2 \quad e_8 = 35\ 0 \quad g_8 = 22\ 0$$

By the same methods (see Fig 42) —

$$c_3 = 53\ 6 \quad e_3 = 65\ 5 \quad g_3 = 41\ 7$$
$$g_7 = 53\ 6 \quad e_7 = 65\ 5 \quad c_7 = 41\ 7$$
$$c_4 = 65\ 2 \quad e_4 = 87\ 1 \quad g_4 = 57\ 0$$
$$g_6 = 65\ 2 \quad e_6 = 87\ 1 \quad c_6 = 57\ 0$$
$$c_5 = 65\ 5 \quad e_5 = 95\ 3 \quad g_5 = 65\ 5$$

These values have been computed with rather less trouble and with greater accuracy by the formulæ for the deflection at any point in a beam due to a load at any other point (see Morley s Strength of Materials) The graphical method of determining the constants should not be relied upon except as a check as small errors in the constants give rise to large errors in the calculated reactions

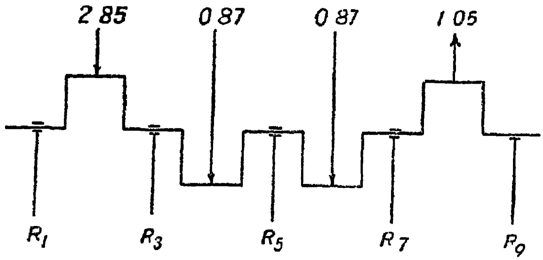


FIG 43

The conditions of loading will now be considered

Case I Crank 1 on Firing Dead Centre —The magnitudes and directions of the applied forces are shewn in Fig 43 Hence —

$$R_2 c_2 = 2\ 85 \times 30\ 2 = 86\ 1 \qquad R_4 c_4 = 0\ 87 \times 65\ 2 = 56\ 7$$
$$R_2 e_2 = 2\ 85 \times 35\ 0 = 99\ 7 \qquad R_4 e_4 = 0\ 87 \times 87\ 1 = 75\ 8$$
$$R_2 g_2 = 2\ 85 \times 22\ 0 = 62\ 7 \qquad R_4 g_4 = 0\ 87 \times 57\ 0 = 49\ 6$$

$$R_6 c_6 = 0\ 87 \times 57\ 0 = 49\ 6 \qquad R_8 c_8 = -1\ 05 \times 22\ 0 = -23\ 1$$
$$R_6 e_6 = 0\ 87 \times 87\ 1 = 75\ 8 \qquad R_8 e_8 = -1\ 05 \times 35\ 0 = -36\ 8$$
$$R_6 g_6 = 0\ 87 \times 65\ 2 = 56\ 7 \qquad R_8 g_8 = -1\ 05 \times 30\ 2 = -31\ 7$$

From which

$$R_2 c_2 + R_4 c_4 + R_6 c_6 + R_8 c_8 = 169\ 3$$
$$R_2 e_2 + R_4 e_4 + R_6 e_6 + R_8 e_8 = 214\ 5$$
$$R_2 g_2 + R_4 g_4 + R_6 g_6 + R_8 g_8 = 137\ 3$$

Substituting these values in equations (1) (2) and (3) we obtain —

$$53\ 6\ R_3 + 65\ 5\ R_5 + 41\ 7\ R_7 = -169\ 3 \qquad (4)$$
$$65\ 5\ R_3 + 95\ 3\ R_5 + 65\ 5\ R_7 = -214\ 5 \qquad (5)$$
$$41\ 7\ R_3 + 65\ 5\ R_5 + 53\ 6\ R_7 = -137\ 3 \qquad (6)$$

From which

$$R_3 = -2\ 434 \quad R_5 = -0\ 735 \quad R_7 = +0\ 230$$

Equating moments about A —

$$8\ R_9 + 7\ R_8 + 6\ R_7 + 5\ R_6 + 4\ R_5 + 3\ R_4 + 2\ P_3 + R\ \ = 0$$

$$8\ R_9 = 7 \times 1\ 05 - 6 \times 0\ 230 - 5 \times 0\ 87 + 4 \times 0\ 735 - 3 \times 0\ 87 + 2 \times 2\ 434 - 2\ 850$$

Whence $R_9 = 0\ 496$

Equating moments about I —

$$8\ R_1 + 7\ R_2 + 6\ R_3 + 5\ R_4 + 4\ R_5 + 3\ R_6 + 2\ R_7 + R_8 = 0$$

$$8\ R_1 = -7 \times 2\ 85 + 6 \times 2\ 434 - 5 \times 0\ 87 + 4 \times 0\ 735 - 3 \times 0\ 87 - 2 \times 0\ 230 + 1\ 05$$

Whence $R_1 = -1\ 097$

Knowing all the forces the bending moment at A B C etc can now be tabulated thus —

Point	Moments	B M n in lb
A		0
B	$10\ 970 \times 10$	+109 700
C	$10\ 970 \times 20 - 28\ 500 \times 10$	−65 600
D	$10\ 970 \times 30 - 28\ 500 \times 20 + 4\ 340 \times 10$	+2 500
E	$10\ 970 \times 40 - 28\ 500 \times 30 + 4\ 340 \times 20 - 8\ 700 \times 10$	−16 400
F	$-4960 \times 30 + 10\ 500 \times 20 - 2300 \times 10$	+38 200
G	$-4960 \times 20 + 10\ 500 \times 10$	−5 800
H	-4960×10	−49 600
I		0

Since the bending modulus of the shaft is

$$\frac{\pi}{32} \times 5\ 25^3 = 14\ 2 \text{ in }^3$$

Therefore

Maximum stress due to bending (occurring in No 1 crank pin at top firing centre) is equal to —

$$\frac{109\ 700}{14\ 2} = 7\ 720 \text{ lb per in }^2$$

Bending stress according to fixed journal method —

$$\frac{W\ L}{8\ Z} = \frac{28\ 500 \times 20}{8 \times 14\ 2} - 5\ 020 \text{ lb sq in}$$

According to free journal method —

$$\text{Stress} = \frac{28\ 500 \times 20}{4 \times 14\ 2} = 10\ 040 \text{ lb sq in}$$

Case II No 2 Crank on Firing Dead Centre —The magnitudes and directions of the applied forces are shewn in Fig 44

$$R_2 = 0\ 870 \quad R_4 = 2\ 850 \quad R_6 = -1\ 05 \quad R_8 = 0\ 870$$

$$R_2 c_2 = 0\ 87 \times 30\ 2 = 26\ 3 \qquad R_4 c_4 = 2\ 85 \times 65\ 2 = 185\ 6$$
$$R_2 e_2 = 0\ 87 \times 35\ 0 = 30\ 5 \qquad R_4 e_4 = 2\ 85 \times 87\ 1 = 248\ 0$$
$$R_2 g_2 = 0\ 87 \times 22\ 0 = 19\ 2 \qquad R_4 g_4 = 2\ 85 \times 57\ 0 = 162\ 3$$

$$R_6 c_6 = -1\ 05 \times 57\ 0 = -59\ 8 \qquad R_8 c_8 = 0\ 87 \times 22\ 0 = 19\ 2$$
$$R_6 e_6 = -1\ 05 \times 87\ 1 = -91\ 4 \qquad R_8 e_8 = 0\ 87 \times 35\ 0 = 30\ 5$$
$$R_6 g_6 = -1\ 05 \times 65\ 2 = -68\ 5 \qquad R_8 g_8 = 0\ 87 \times 30\ 2 = 26\ 3$$

From which

$$R_2 c_2 + R_4 c_4 + R_6 c_6 + R_8 c_8 = 171\ 3$$
$$R_2 e_2 + R_4 e_4 + R_6 e_6 + R_8 e_8 = 217\ 6$$
$$R_2 g_2 + R_4 g_4 + R_6 g_6 + R_8 g_8 = 139\ 3$$

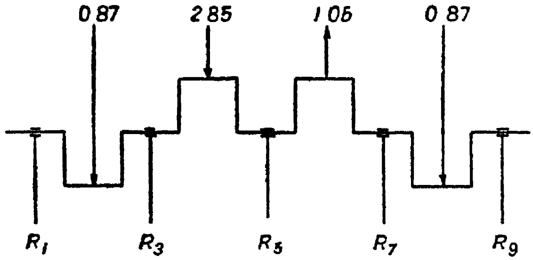


Fig 44

The three equations for determining $R_7 R_5 R_3$ are therefore —

$$53\ 6\ R_7 + 65\ 5\ R_5 + 41\ 7\ R_3 = -171\ 3$$
$$65\ 5\ R_7 + 95\ 3\ R_5 + 65\ 5\ R_3 = -217\ 6$$
$$41\ 7\ R_7 + 65\ 5\ R_5 + 53\ 6\ R_3 = -139\ 3$$

from which

$$R_7 = -2\ 514 \quad R_5 = -0\ 713 \quad R_3 = 0\ 226$$

Taking moments about A and I

$$R_1 = -0\ 072 \quad R_9 = -0\ 467$$

from which the following figures for the bending moments are obtained —

Point	Moments	B M in in lb
A		0
B	720×10	+7 200
C	$720 \times 20 - 8700 \times 10$	−72 600
D	$720 \times 30 - 8700 \times 20 + 25\ 140 \times 10$	+99 000
E	$720 \times 40 - 8700 \times 30 + 25\ 140 \times 20 - 28\ 500 \times 10$	−14 400
F	$4670 \times 30 - 8700 \times 20 - 2260 \times 10$	−56 500
G	$4670 \times 20 - 8700 \times 10$	+6 400
H	4670×10	+46 700
I		0

$$\text{Maximum bending stress } \frac{99\ 000}{14\ 2} = 6\ 970 \text{ lb sq in}$$

Considerably less than in Case I

Case III Loading as in Case I but bearings C and G supposed worn down 20 thousandths and bearing E 25 thousandths of an inch below the level of the line joining A I

The difference in height of the bearings is very simply allowed for by putting 20 25 and 20 respectively on the right hand side of equations (1) (2) and (3) instead of zero

The equations for determining R_3 R_5 and R_7 then become —

$$53\ 6\ R_3 + 65\ 5\ R_5 + 41\ 7\ R_7 = 20 - 169\ 3 = -149\ 3$$

$$65\ 5\ R_3 + 95\ 3\ R_5 + 65\ 5\ R_7 = 25 - 214\ 5 = -189\ 5$$

$$41\ 7\ R_3 + 65\ 5\ R_5 + 53\ 6\ R_7 = 20 - 137\ 3 = -117\ 3$$

The following values are obtained for the reactions at the bearings —

$$R_1 = -1\ 398 \quad R_3 = -1\ 880 \quad R_5 = -1\ 243 \quad R_7 = +0\ 790$$
$$R_9 = +0\ 191$$

The bending moments are as follows —

Point	Moments	B M in in lb
A		0
B	13 980 × 10	+139 800
C	13 980 × 20 − 28 500 × 10	−5 400
D	13 980 × 30 − 28 500 × 20 + 18 800 × 10	+37 400
E	13 980 × 40 − 28 500 × 30 + 18 800 × 20 − 8700 × 10	−6 800
F	1910 × 30 + 10 500 × 20 − 7 900 × 10	+73 700
G	1910 × 20 + 10 500 × 10	+66 800
H	1910 × 10	+19 100
I		0

Maximum bending stress $\frac{139\ 800}{14\ 2} = 9\ 800$ lb sq in

Comparing the above figures with those obtained in Case I it will be seen that the maximum stresses have been increased to the extent of about 28% by the difference of level of the bearings In view of the uncertainty which exists with regard to the deflection of cranked shafts the above figures are not strictly reliable but the writer is of opinion that they under rather than over estimate the stresses

Conclusions —1 The value of the bending moments at a crank pin on the top firing centre is greater for a crank pin situate at one end of the shaft than that at one nearer the centre of the shaft

2 The bending moments at certain crank pins and journals may be as great or greater than the bending moment at the crank pin which is receiving the greatest applied load

3 No general rule for the bending moment at a Diesel Engine crank pin or journal can represent the true state of affairs but every different arrangement of cranks and number of cylinders requires to be investigated individually

4 Difference of level of the bearings due to wear or other wise gives rise to greatly increased bending moments which can be calculated approximately in the manner described

The methods of calculation which have been illustrated in

this chapter can be applied to cases involving any number of cranks and the effects of fly wheels the rotors of electric generators outboard bearings etc can be included When the number of cylinders is three or less the weight of the fly wheel is considerable and cannot therefore be ignored In these cases allowance should be also made for the practice of packing the outward bearing above the level of the engine main bearings For engines of four cylinders and over the weight of the fly wheel and the presence of outboard bearings can probably be neglected with safety An extended treatment of this subject will have to be reserved for a future occasion

The labour involved in solving simultaneous linear equations increases as the square of the number of unknowns For a description of a machine devised to do this work mechanically see the Treatise on Natural Philosophy vol i Kelvin and Tait

Graphical Determination of the Twisting Moments —In the processes described below the following approximations have been made —

1 The negative twisting moments due to the air compressor at the forward end of the engine have been neglected These moments are small in comparison with the moments due to the working cylinders and being opposite in direction to the maximum moments tend to reduce the latter by a small amount

2 Moments due to the dead weight of the revolving and reciprocating parts have also been neglected In a very large engine it would be advisable to take these into consideration as other things being equal the dead weight of the running gear per square inch of piston area increases as the scale of the engine

3 The twisting moments due to mechanical friction have been neglected as (so far as the present writer is aware) the distribution and variation of the friction forces are not known with any exactitude and in any case one is a little on the safe side in neglecting them These friction moments of course accumulate as one passes from the forward to the aft end of the engine where they amount in aggregate to about 15% of the mean indicated twisting moment so their effect on the forward end of the shaft is quite negligible

4 The moment of inertia of the fly wheel has been assumed to be large in comparison with the fly wheel effect of the revolving and reciprocating parts of the running gear In cases where the fly wheel is very small or omitted altogether as in some two stroke marine engines the irregularities of turning effort are mainly absorbed by the angular acceleration of the crank masses Allowance is readily made for this effect in the following manner The combined twisting moment curve for all the cylinders is first found without allowance for fly wheel effect and a new zero line is taken at the height corresponding to the mean twisting moment Ordinates measured to this new zero line represent fluctuations of the twisting moment from its mean value These ordinates are now divided into segments proportional to the fly wheel effects of the fly wheel and crank masses For example if there are four cylinders and the moment of inertia of the fly wheel is three times that of one set of crank masses then the ordinates will be divided into seven parts one part being applied in opposite sense to corresponding points on the twisting moment curve of each cylinder and the remaining three parts of each ordinate form the ordinates of a curve of the twisting moments absorbed by the angular inertia of the fly wheel

In the example worked out below it will be assumed that the fly wheel is large compared with the crank masses so that the process described briefly above is not necessary

Fig 45 is a skeleton diagram of the connecting rod positions for every 20 degrees of revolution of the crank shaft for the determination of piston displacements

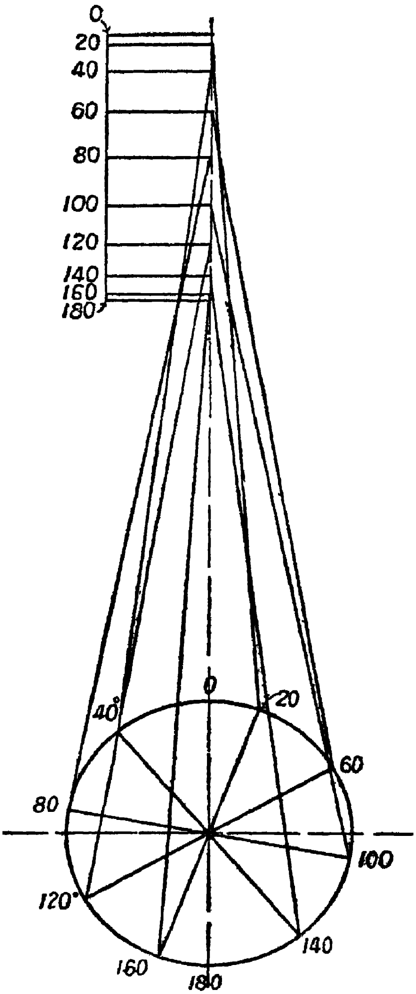


Fig 45

Fig 46 is a typical full load indicator card calibrated for pressures vertically and percentages of stroke horizontally Points corresponding to each 20 degrees of revolution are marked on the diagram by scaling the piston displacements off Fig 45

On Fig 47 cylinder pressures are plotted on a crank angle base from 0 to 720 degrees (four stroke engine) The pressure

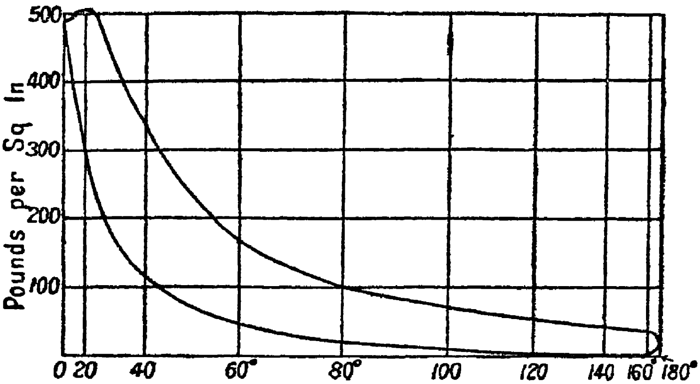


FIG 46

during the suction and exhaust strokes is assumed atmospheric The inertia effect of the reciprocating parts per square inch of piston area is plotted from the following figures —

$$\text{Inertia effect at top dead centre } \frac{236\times 615}{g}(1+\tfrac{1}{8})=5400\text{ lb}$$

$$\text{bottom} \quad \frac{236\times 615}{g}(1-\tfrac{1}{8})=3600$$

$$90 = \frac{236\times 615}{g}\times\frac{1}{5}= 900\text{ lb}$$

$$45 = \frac{236\times 615}{g}\times\frac{1}{\sqrt{2}}=3200\text{ lb}$$

Corresponding figures per sq in of piston area (78 5 sq in) are 69 46 11 5 and 41 lb per sq in respectively

The centrifugal forces of the revolving masses being radial produce no twisting effect on the shaft The dotted line (Fig 47) is the resultant of the pressure and inertia curves The twisting moments are now computed as in the following table —

Degrees from top dead centre	Leverage in ins	Resultant force in lb/in of piston area	Twisting moments in lb	Degrees from top dead centre	Leverage in ins	Resultant force in lb/in of piston area	Twisting moments in lb
0	0	418	0	360	0	—69	0
20	3 00	444	104 000	340	—3 00	—60	14 100
40	5 63	284	125 000	320	—5 63	—46	20 400
60	7 20	138	78 000	300	—7 20	—25	14 100
80	7 65	100	60 000	280	—7 65	0	0
100	7 13	90	50 000	260	—7 13	20	—11 200
120	5 85	90	41 400	240	—5 85	39	—17 900
140	4 08	88	28 200	220	—4 08	47	—15 100
160	2 10	85	14 000	200	—2 10	59	—9 750
180	0	73	0	180	0	73	0
360	0	—69	0	720	0	418	0
380	3 00	—60	—14 100	700	—3 00	232	—54 500
400	5 63	—46	—20 400	680	—5 63	69	—30 500
420	7 20	—25	—14 100	660	—7 20	22	—12 400
440	7 65	0	0	640	—7 65	22	—13 200
460	7 13	20	11 200	620	—7 13	32	—17 900
480	5 85	37	17 000	600	—5 85	42	—19 300
500	4 08	44	14 100	580	—4 08	44	—14 100
520	2 10	45	7 400	560	—2 10	45	—7 400
540	0	46	0	540	0	46	0

The leverage tabulated in the second column is found by the well known graphical construction in Fig 45 where the line of the connecting rod is produced (if necessary) to meet the horizontal line through the centre of the shaft the intercept being the leverage required to the same scale as the rest of the diagram The twisting moments are found by multiplying the leverage in inches by the resultant forces per sq in of

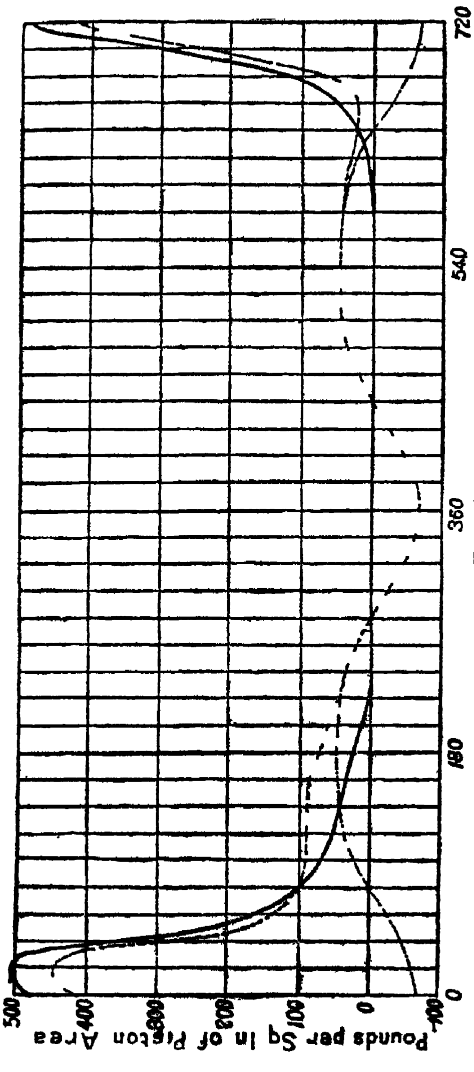


Fig 47

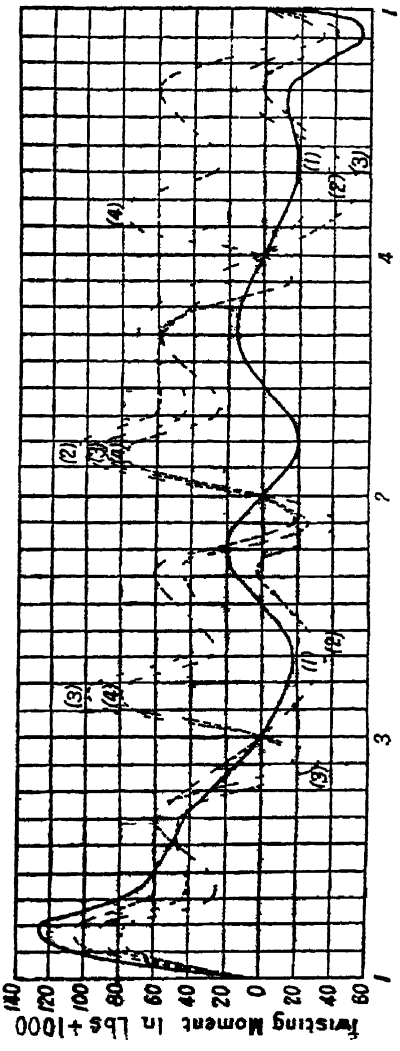


Fig 48

piston area in lb per sq in and by the piston area in sq in (in this case 78 5 sq in)

Forces acting towards the crank are considered positive whether they are expansion forces or otherwise and those acting away from the crank negative Assuming rotation clockwise leverages to the right hand of the centre line are positive and those to the left negative The signs of the moments then look after themselves according to the signs of their factors It is not unusual to see the leverage in a case of this sort treated as though it were always positive The dis advantage of this proceeding is that in order to get the signs of the moments correct those of the resultant forces have to be reversed at every dead centre which besides being incorrect from a mathematical standpoint is inconvenient for the draughtsman and confusing to others The tabulated values of the twisting moment are plotted in Fig 48 (full line curve) Identical curves for cylinders 3 4 and 2 could be plotted in their respective places at 180 degrees apart in the order named but are omitted for the sake of clearness Dotted curve numbered 2 is the resultant of the curves belonging to cylinders 1 and 2 Dotted curve 3 is the resultant of the curves belong ing to cylinders 1 2 and 3 and so on The simplest way of obtaining these resultants is to trace the primary curve on a piece of transparent paper and move it sideways into its required position for the next cylinder and then for every required ordinate move the paper vertically (guided by the vertical degree lines) until the zero line coincides with the top of the ordinate of the curve to which it is required to add the effect of another cylinder In this position prick through the top of the ordinate of the curve on the tracing paper to the diagram underneath These resultant curves enable the twisting moment at any crank pin or journal at any angular position to be read off the diagram

Combined Effect of Bending and Twisting —It will be seen that the peaks of the twisting moment curves occur about 30 degrees after the dead centres and that the results previously obtained for the bending moments with the cranks on dead centre apply very closely to this position also so that the tabulated values of the bending moments at the various journals and crank pins combined with the twisting moments existing at these points 30 degrees after the corresponding firing dead centres have been passed represent the maximum

conditions of stress at the points in question The conditions of bending when cranks 3 and 4 are on firing centre are of course the same as those obtaining when cranks 2 and 1 respectively are in that position the order in which the bending moments occur being reversed

For example the bending moment at No 2 crank pin when No 4 cylinder is firing is the same as the bending moment at No 3 crank pin when No 1 cylinder is firing and so on A comparison of the following table with the twisting moment curves and the bending moments tabulated in the previous articles will make the matter clear

The equivalent twisting moment equals $\sqrt{T^2+B^2}$ and is that twisting moment which would give the same shear stress as the maximum shear stress due to the combined action of twisting and bending moments actually obtaining

Position	Which crank 30 past firing dead centre	Bending moment in lb (see previous tables)	Number of twisting moment curve	Twisting moment in lb from curves	Equivalent twisting moment in lb	Maximum shear stress lb per sq in
A Journal	—	—	—	—	—	—
B Crank pin No 1	No 1 No 2 No 3 No 4	109 700 7 200 46 700 49 600	—	—	109 700	3 870
C Journal	No 1 No 2 No 3 No 4	65 600 72 600 6 400 5 800	(1)	127 000 11 000 13 000 17 000	143 000	5 040
D Crank pin No 2	No 1 No 2 No 3 No 4	2 500 99 000 56 500 38 200	(1)	127 000 11 000 13 000 17 000	127 000	4 480
E Journal	No 1 No 2 No 3 No 4	16 400 14 400 14 400 16 400	(2)	112 000 115 000 31 000 31 000	116 000	4 090

Position	Which crank 30 past firing dead centre	Bending moment in lb (see previous tables)	Number of twisting moment curve	Twisting moment in lb from curves	Equivalent twisting moment in lb	Maximum shear stress lb per sq in
F Crank	No 1	38 200		112 000		
pin	No 2	56 500	(2)	115 000	128 000	4 510
No 3	No 3	99 000		31 000		
	No 4	2 500		31 000		
	No 1	5 800		100 000		
G Journal	No 2	6 400	(3)	93 000	100 000	3 530
	No 3	72 600		93 000		
	No 4	65 600		45 000		
H Crank	No 1	49 600		100 000		
pin	No 2	46 700	(3)	93 000	112 000	3 950
No 4	No 3	7 200		93 000		
	No 4	109 700		45 000		
	No 1			82 000		2 890
I Journal	No 2	—	(4)	82 000	—	
	No 3			82 000		
	No 4			82 000		

Conclusions —Maximum Shear Stress 5 040 lb sq in — Taking the fatigue stress in shear for mild steel subject to combined bending and twisting at 15 000 lb per sq in the factor of safety for a shaft newly lined up is about 3 and diminishes very considerably as the bearings become worn out of level

The high values of the stresses at the centre of the shaft point to the advisability of making all couplings between sections of the crank shaft of the full torsional strength of the shaft i e the aggregate shearing area of the coupling bolts multiplied by the radius of their pitch circle should be equal to the twisting modulus of the shaft Thickness of coupling flanges $^1/_4$ diameter of the shaft

CHAPTER VI

FLY WHEELS

The Functions of a Fly wheel are —

1 To keep the degree of uniformity within specified limits

2 Where alternators running in parallel are in question to limit the angular advance or retardation of rotation to a specified fraction of a degree ahead of or behind an imaginary engine rotating with perfectly uniform angular speed

3 To limit the momentary rise or fall in speed when full load is suddenly thrown off or on

4 To facilitate starting under compressed air

In addition to the above the fly wheel usually serves as a barring or turning wheel and a valve setting disc also the inertia of the fly wheel has great influence in determining the critical speed at which torsional oscillations of the crank shaft are set up

Fly wheel Effect —The fly wheel effect of a rotating body is its polar moment of inertia (mass $\times$ radius of gyration squared) about its axis of rotation For a fly wheel or pulley it is found approximately by multiplying the weight of the rim in pounds by the square of the distance in inches from the axis to the centre of gravity of the section of the rim the result being in in.^{2} lb units This underestimates the moment of inertia slightly and a more accurate method will be described later The fly wheel effect of the running gear of one cylinder is found with sufficient accuracy for most purposes by adding the weight of the revolving parts (crank pin plus unbalanced part of two crank webs plus 0 65 of the connecting rod) to half the weight of the reciprocating parts (0 35 of the connecting rod plus cross head plus piston rod plus piston etc) and multiplying the sum by the square of the crank radius

For a screw propeller the radius of gyration may be taken as 0 35 of the extreme radius if details are not available

Degree of Uniformity —

$$\text{Degree of uniformity} = \frac{\text{Max speed} - \text{Min speed}}{\text{Mean speed}}$$

Let d = Degree of uniformity

w_1 = Max angular speed in radians per second

w_2 = Min

$$\text{Then } d = 2\frac{(w_1 - w\ )}{(w_1 + w_2)} \qquad (1)$$

For a specified value of d the necessary fly wheel effect is calculated by means of the resultant twisting moment curve of the engine Let Fig 49 represent the twisting moment curve and the line A C the mean twisting moment Let A B C be the loop of largest area (with multi cylinder engines there are in general as many positive and negative loops in a complete cycle as there are cylinders and the area of each positive loop is the same as that of each negative loop) If the loop A B C is above the line A C then the speed of the engine is a minimum at A and a maximum at C and the increase of rotational energy of the fly wheel etc between A and C is equal to the work represented by the area of the loop A B C

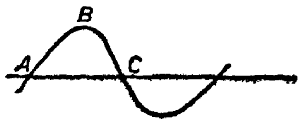


FIG 49

Let A = Area of loop A B C in sq in on the diagram

E = Work represented by A B C in in lb

a = Scale to which turning moments are plotted in in lb to the inch

b = Scale to which crank shaft degrees are plotted in degrees to the inch

$$\text{Then } E = \frac{A \times a \times b}{57\ 3}$$ 57 3 being the number of degrees in a radian

Let WK^2 = Fly wheel effect (moment of inertia) in in ^{2} lb

$$\text{Then kinetic energy of wheel} = \frac{WK^2\ w^2}{2\ g} \quad (g = 386 \text{ in } /\text{sec }^2)$$

Change of kinetic energy from A to C

$$= \frac{WK^2}{2\ g}(w_1{}^2 - w_2{}^2) = \frac{WK^2}{2\ g}(w_1 - w_2)\ (w_1 + w_2) = \frac{WK^2\ d}{4\ g}(w_1 + w_2)^2$$

$$= WK^2\ d\ w^2(\text{MEAN}) - g$$

if the difference between w_1 and w_2 is small

But the change of kinetic energy is equal to E

$$E=\frac{WK^2}{386}\ d\ w^2$$

and $d=\dfrac{E\times 386}{WK^2\ w^2}$ or $WK^2=\dfrac{E\times 386}{w^2\ d}$ (2)

Example Single cylinder engine 10 bore × 15 stroke Revs 300 Turning moment diagram as in Fig 48 full line E=151 000 in lb Radius of gyration of wheel 30 Required to find the weight of the wheel to give a degree of uniformity of 1/80

$$w=\frac{300\times 2\pi}{60}=31\ 4 \text{ radians per sec}$$

$$W\ K^2=\frac{E\times 386}{w^2\ d}=\frac{151\ 000\times 80\times 386}{31\ 4^2}=4\ 730\ 000$$

Fly wheel effect of running gear $\left(267+\frac{236}{2}\right)\times 7\ 5^2=21\ 600$ in ^{2} lb

WK^2 for fly wheel $=4\ 730\ 000-21\ 600=4\ 708\ 400$ in ^{2} lb

but $K=30$

$$W=\frac{4\ 708\ 400}{30^2}=5230 \text{ lb } =2\ 34 \text{ tons}$$

Twisting Moment Diagrams for two and four stroke engines having from one to eight cylinders are shewn in Figs 50 to 61 These have been drawn for an engine 10 bore by 15 stroke As the twisting moments of two engines of different sizes are proportional to the bore^2 × stroke these curves may be used for engines of any size by multiplying the moments by the bore^2 (in inches^2) × the stroke (in inches) and dividing by 1500 The excess energy represented by the largest loop in each diagram is given in the schedule below for each case

Number of Cylinders	Four Stroke Engines: E in in lb for 10 × 15 Cylinder	Four Stroke Engines: E in in lb for 1 × 1 Cylinder	Two Stroke Engines: E in in lb for 10 × 15 Cylinder	Two Stroke Engines: E in in lb for 1 × 1 Cylinder
1	151 000	101 0	125 500	83 7
2	127 500	84 8	58 500	39 0
3	87 300	58 2	48 700	32 4
4	38 500	25 7	39 000	26 0
6	39 100	26 1	11 150	7 4
8	31 700	21 1	2 200	1 5

Substituting those values of E for a cylinder 1 in $\times$1 in in equation (2) the following formula is obtained —

$$WK^2 = \frac{C\,B^2\,S}{d\left(\frac{n}{100}\right)^2}$$

Where B =Bore of cylinder in inches
S =Stroke in inches
n =Revolutions per minute

Values of C are given in the following schedule —

No of Cylinders	C for 4 Stroke Engine	C for 2 Stroke Engine
1	355	243
2	298	137
3	204	114
4	90	91
6	91	26
8	74	5

Values for d used in Diesel Engine Practice —For certain purposes as for instance spinning mills a fine degree of uniformity is desirable and d=about $\frac{1}{100}$

For direct coupled continuous current dynamos d=$\frac{1}{80}$ is sufficiently fine to prevent flickering of lights and may be used unless considerations of momentary governing demand a heavier wheel than the use of this figure would give rise to

For marine engines and land drives where regularity of turning is not of importance d may be about $\frac{}{40}$

The above values for the degree of uniformity must be used with caution as in a large number of cases (particularly four stroke engines of six cylinders and upwards and two stroke engines of three cylinders and upwards) the considerations discussed in the next article outweigh those of regularity in turning

Momentary Governing —Under the head of governing it is usually specified that the rise in speed when the load is thrown off suddenly or the fall in speed when the load is suddenly thrown on shall not exceed a certain percentage (usually between 5 and 12) of the mean speed Actually the governor

has relatively small control over this rise or fall of speed as at the instant when the load is thrown off sufficient fuel has already been deposited in the pulverisers to carry the engine against full load for a period which may be anything up to

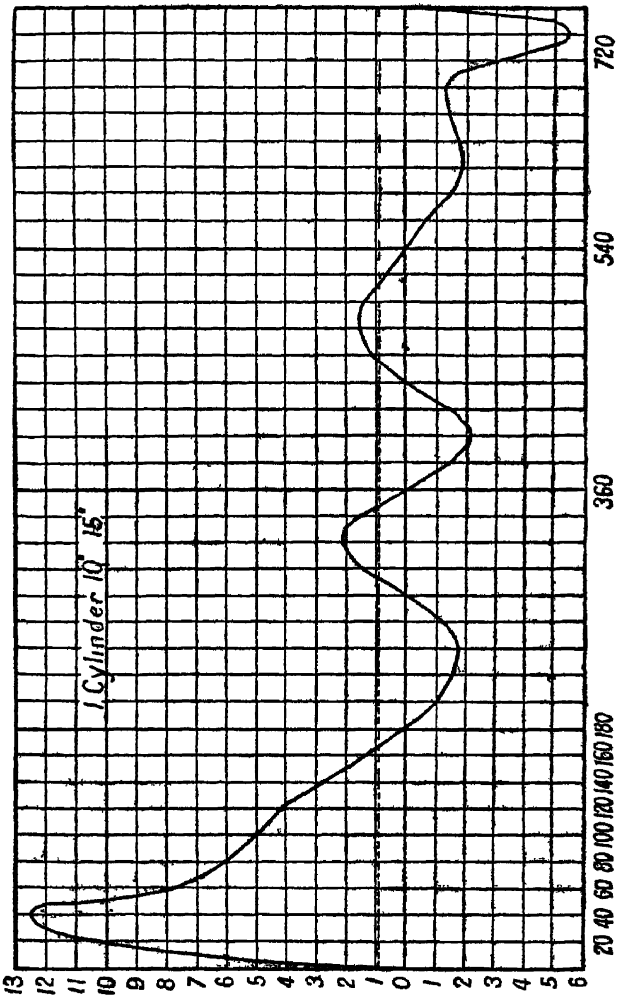


Fig 50

FOUR STROKE ENGINE

two revolutions in the case of a four stroke engine The brake energy developed during this period is entirely devoted to accelerating the fly wheel and other rotating masses Owing to the fact that the governor does not act immediately the load is thrown off the wheel should be capable of absorbing the

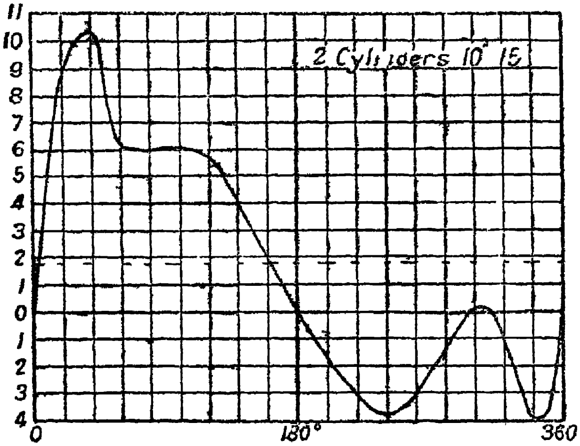


FIG 51

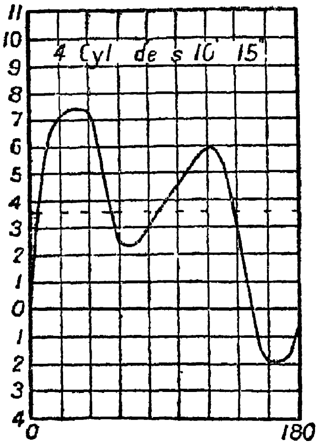


FIG 52

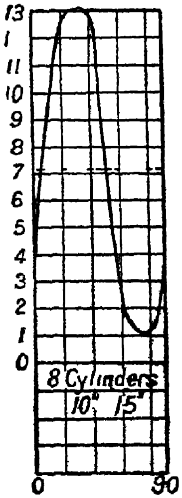


FIG 53

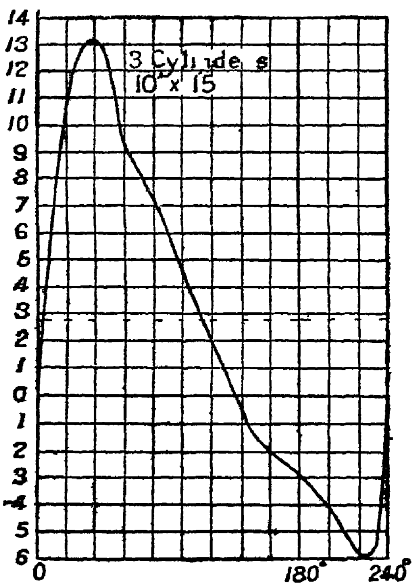


FIG 54

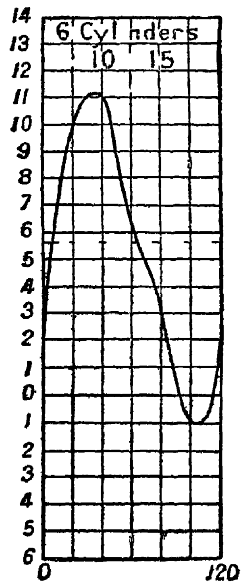


FIG 55

FOUR STROKE ENGINES

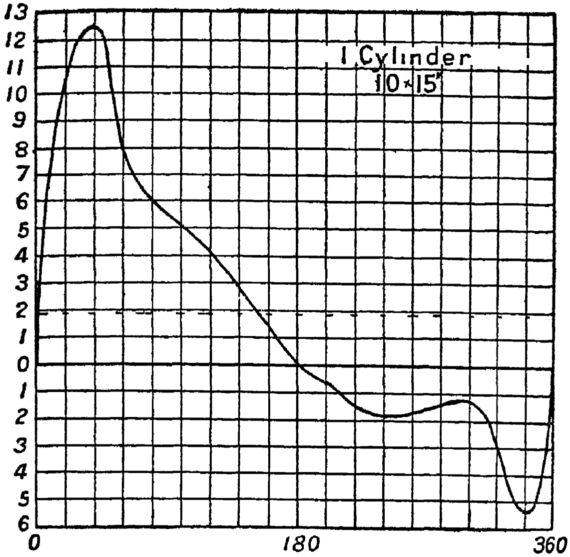


Fig 56

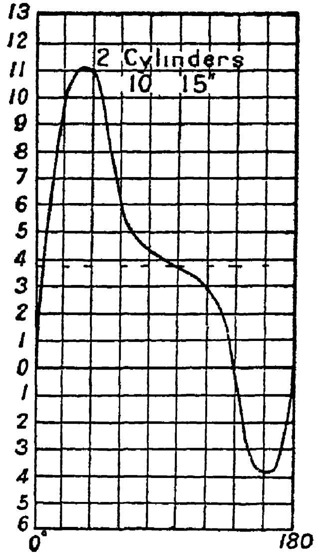


Fig 57

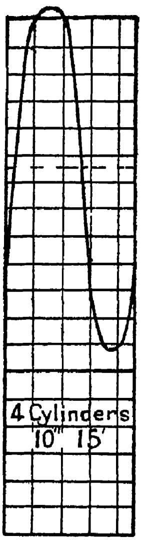


Fig 58

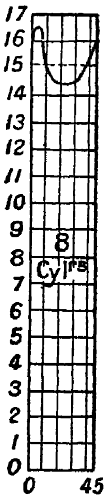


Fig 59

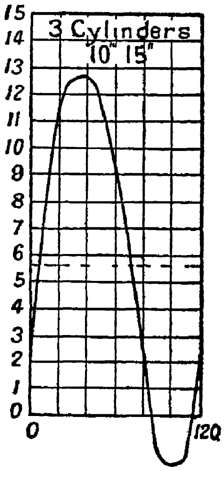


Fig 60

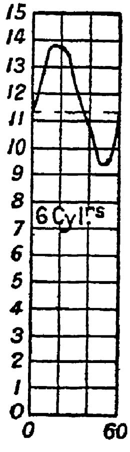


Fig 61

TWO STROKE ENGINES

whole power of the engine for about three revolutions in the case of a four stroke engine about 1 5 of a revolution in the case of a two stroke engine

Example	B H P of engine (four stroke)	180
	Revolutions per minute	375
	Momentary rise in speed when full load is suddenly thrown off	12%
	Radius of gyration of wheel	18 in

It is required to find the weight of the fly wheel neglecting the fly wheel effect of the running gear

Work done per revolution at full load $\dfrac{180 \times 33\ 000 \times 12}{375}$ in lb

Energy corresponding to three revolutions

$$=\frac{180 \times 33\ 000 \times 36}{375} = 570\ 000 \text{ in lb}$$

Angular speed at full load $\dfrac{2\pi \times 375}{60} = 39\ 3$ radians per second

Momentary angular speed when load is suddenly thrown off

$$39\ 3 \times 1\ 12 = 44\ 0 \text{ radians per second}$$

If W=weight of wheel then —

$$\frac{W \times 18^2}{2 \times 386}(44^2 - 39\ 3^2) = 570\ 000 \text{ in }^2\text{ lb}$$

and W=3470 lb

Alternators in Parallel —When two or more alternator sets are being run in parallel it is a necessary condition for working that they keep almost exactly in phase Due to inequalities of twisting moment slight differences of phase inevitably occur and these give rise to synchronising currents between the various machines the tendency of these currents being to accelerate the lagging machines and retard the leading ones This effect keeps the whole system in a state of stability but cannot be relied on to correct any large fluctuations and on this account it is usual to specify that the maximum deviation from uniform rotation shall not exceed three electrical degrees on either side of the mean If the alternator under consideration has a field of two poles only then the electrical degrees correspond to crank shaft degrees In general if the number of pole pairs is p then one crank shaft degree corresponds

to p electrical degrees So far as the engine designer is concerned then the problem consists in ascertaining the fly wheel effect required to keep the cyclic fluctuations on th engine fly wheel within a certain number of degrees or more commonly within a certain fraction of a degree of revolution on either side of the mean

The method of calculation may be described briefly thus —

(1) Assume any convenient figure foi the fly wheel effect e g 100 000 in ^{2} lb

(2) Plot twisting moment curve for complete period taking the zero of ordinates at the mean twisting moment

(3) Reduce crank angles to time in seconds assuming uniform rotation

(4) Reduce twisting moments to angular acceleration in degrees per second^2 by dividing by the assumed fly wheel effect and by the acceleration due to gravity (386 in per sec ^{2}) and multiplying by the number of degrees in a radian (57 3)

(5) Plot angular acceleration to time or crank angle base

(6) Integrate by planimeter or otherwise obtaining angular speed curve

(7) Integrate again obtaining angular displacement curve

(8) Measure maximum deviation from the mean position in degrees

(9) Increase or decrease the assumed fly wheel effect in pro portion as the angular deviation so found is more or less than the deviation specified This gives the fly wheel effect required

Example Three cylinder four stroke engine —

Bore	20 in
Stroke	32
Revolutions per minute	150
Number of pole pairs	20
Angular deviation	3 electrical deg
Twisting moment curve as in Fig 62 The mean twisting moment being taken as the basis	

The whole calculation is contained in the table below in con junction with Figs 62 63 and 64

Since the engine makes 150 revolutions per minute therefore

$$20^\circ = \frac{20 \times 60}{150 \times 360} = 0\ 0222 \text{ sec}$$

Assume fly wheel effect of 10^6 in ^{2} lb for purposes of calcula tion Then —

$$\text{Acceleration in radians per sec}^2 = \frac{\text{Twisting moment} \times 386}{10^6}$$

$$\text{in degrees per sec}^2 = \frac{\text{T M} \times 57\ 3 \times 386}{10^6} = \frac{\text{T M}}{45\ 3}$$

Referring to the table below —

Values given in column 2 are scaled off Fig 62

3 are obtained by dividing those in column 2 by 45 3

4 are obtained by multiplying the values in column 3 by 0 0222 sec

Column 5 is obtained by successive addition of speed increments

Column 6 contains corrections necessitated by the fact that the resultant of column 5 is not zero owing to errors

Column 7 gives corrected speeds which are plotted in Fig 63

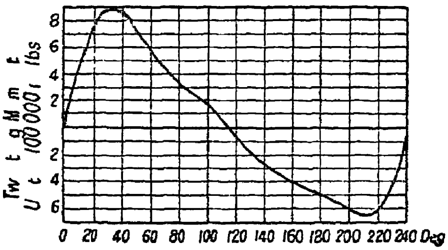


FIG 62

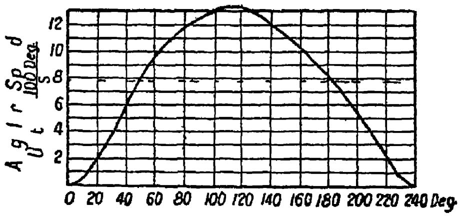


FIG 63

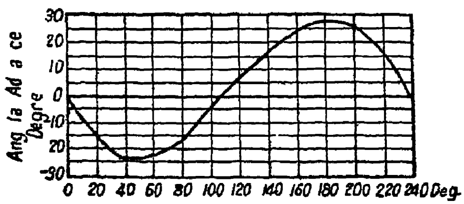


FIG 64

Columns 8 9 and 10 are obtained similarly to columns 2 4 and 5

Total swing in phase (see Fig 64) $24 + 28 = 52°$ or 26 each side of the mean

Degrees	Average Twisting moment over 20° In.-lb. ÷ 1000	Average Acceleration over 20° Deg./sec.	Increase or Decrease in Speed Deg./sec.	Summation of Speed Increments Deg./sec.	Correction Deg./sec.	Corrected Speed Deg./sec.	Average Speed above or below Mean Speed Deg./sec.	Increase or Decrease of Angle Deg.	Summation of Angular Advances or Retardations Deg.
0				0	0	0			0
	440	9710	216				—700	—15.5	
20				216	9	207			—15.5
	870	19 200	426				—340	—7.5	
40				642	18	624			—23.0
	730	16 200	360				40	0.9	
60				1002	27	975			—22.1
	440	9710	216				300	6.7	
80				1218	35	1183			—15.4
	270	5960	132				480	10.7	
100				1350	44	1306			—4.7
	30	662	15				530	11.8	
120				1365	53	1310			7.1
	—180	—3980	—88				470	10.4	
140				1277	62	1215			17.5
	—340	—7500	—167				330	7.3	
160				1110	71	1039			24.8
	—430	—9480	—211				150	3.3	
180				899	80	819			28.1
	—540	—11 900	—264				—90	—2.0	
200				635	88	547			26.1
	—640	—14 100	—313				—390	—8.7	
220				322	97	225			17.4
	—440	—9710	—216				—700	—15.5	
240									1.9

Allowable swing 3 electrical deg $=\frac{3}{20}=$crank shaft deg

Fly wheel effect assumed for calculation 1 000 000 in ^{2} lb

Therefore fly wheel effect required

$$=\frac{1\ 000\ 000\times 20\times 26}{3}=173\times 10^6 \text{ in}^2 \text{ lb}$$

If radius of gyration of wheel is 65 in

$$\text{Weight of wheel}=\frac{173\times 10^6}{65^2\times 2240}=18\ 3 \text{ tons}$$

Allowance for the fly wheel effect of the alternator rotor would reduce this figure a little

Torsional Oscillations and Critical Speeds —In the great majority of land Diesel Engines the critical speed at which torsional oscillations of the crank shaft would occur lies far above the practical range of the engine With marine installations the probability of a critical speed occurring within the working range of speed is higher In any case the possibility should be examined and fortunately the trouble can usually be avoided by a suitable modification to the fly wheel or the shafting Supposing for the moment that a marine engine is under consideration Then the fly wheel the crank masses the propeller and the shafting etc constitute an elastic system having a natural frequency of torsional oscillations which depends on the amounts and positions of the fly wheel effects of its component parts and on the stiffness of the shafting If this natural frequency happens to be the same as the frequency of the torsional impulses due to the working strokes of the engine then the oscillations tend to become accumulative vibrations are felt in the shafting and the crank shaft may hammer in its bearings Then the engine is running at a critical speed In general a two stroke engine gives as many torsional impulses per revolution as there are cylinders and a four stroke engine half this number so that a two stroke engine attains its critical speed at one half the number of revolutions required by a four stroke engine For example Suppose the crank shaft etc of a six cylinder four stroke engine has a natural frequency of 2400 complete oscillations per minute then the critical speed will be 800 revolutions per minute A similar engine working on the two stroke cycle would have a critical speed of 400 revolutions per minute It will be obvious that all the revolving masses in connection with

the crank shaft (apart from trifling items) must be taken into consideration so that the critical speed of a marine engine coupled to a dynamo for testing purposes will be different to that obtained when the engine is installed in the ship

Natural Frequency of Torsional Oscillation —Consider the simple system shewn in Fig 65 consisting of a shaft fixed at one end and carrying a fly wheel at the other If the fly wheel be turned through an angle against the torsional resistance of the shaft and then released suddenly the system will oscillate until the energy has been dissipated in friction the angle through which any section of the shaft oscillates being proportional to the distance from the fixed end The latter is called the Node

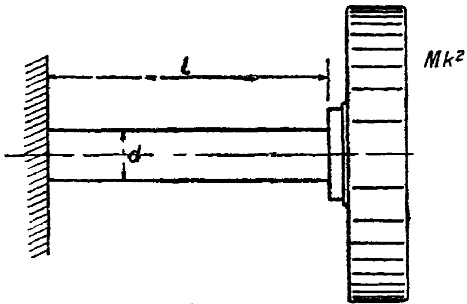


FIG 65

Let f = Frequency in complete oscillations per second
F = Frequency minute
d = Diameter of the shaft in inches
I = Polar moment of inertia of shaft section in inches^4
l = Length of shaft in inches
W = Weight of the wheel in lb
K = Radius of gyration of the wheel in inches
g = Acceleration due to gravity in inches per second^2 (386)
G = Modulus of rigidity of the shaft material (about 12 000 000 lb per sq in)

Then —

$$f = \frac{1}{2\pi}\sqrt{\frac{I\,G\,g}{l\,W\,K^2}} \text{ and } F = 9\ 55\sqrt{\frac{I\,G\,g}{l\,W\,K^2}} \qquad (1)$$

If the shaft is not of the same diameter throughout its length but consists of sections of length l_1 l_2 etc of diameter d_1 d_2 etc then the equivalent length l_0 of shaft of standard diameter d_0 is given by

$$l_0 = l_1\left(\frac{d_0}{d_1}\right)^4 + l_2\left(\frac{d_0}{d_2}\right)^4 + \quad \text{etc}$$

For example 1 ft of 6 in shafting is equivalent to 16 ft of 12 in shafting 16 ft being $\left(\frac{12}{6}\right)^4$ For this reason the lengths occupied by coupling flanges etc , are negligible

If there are a number of fly wheels or other rotating masses at different distances from the node then the frequency is given very closely by the following —

$$F=9\ 55\sqrt{\frac{I\,G\,g}{l_1W_1K_1{}^2+l_2W_2K_2{}^2+\quad \text{etc}}} \qquad (2)$$

In the case of an engine and shafting no point on the latter is fixed and the position of the node is to be inferred from considerations of dynamic equilibrium It will be obvious that a uniform shaft with an equal wheel at each end will have its node at the centre the oscillations of the two wheels being equal in magnitude and opposite in sense at every instant It is almost equally apparent that in a similar case with unequal wheels the position of the node will divide the shaft into two lengths in inverse ratio to the fly wheel effects at their ends

In general the position of the node is determined with sufficient accuracy to enable the critical speed to be predicted within a few revolutions per minute by treating the fly wheel effects (moments of inertia) as though they were weights and locating the node at their centre of gravity

A crank shaft may be treated as a uniform shaft of the diameter of the journals

Example Six cylinder two stroke marine engine

Bore	24 in
Stroke	35 in
Revolutions per minute	120
Diameter of shaft	15 in
Lengths of crank shaft	as in Fig 66
Weight of fly wheel	12 000 lb
Radius of gyration of fly wheel	40 in
Weight of revolving parts for one crank	4 500 lb
reciprocating parts	5 000 lb
propeller	10 000 lb
Radius of gyration of propeller	20 in
Shafting	as in Fig 66

Reduction of Shafting to Standard Diameter of 15 Inches —

Length of 14 shafting 620 in

Equivalent length of 15 shafting $620\left(\frac{15}{14}\right)^4=815$ in

Equivalent length of 15" shafting between the propeller and fly wheel $= 150 + 815 + 60 = 1025$ in

Moment of inertia (polar) of 15" shaft $= \frac{\pi}{32} \times 15^4 = 4960$ in ^{4}

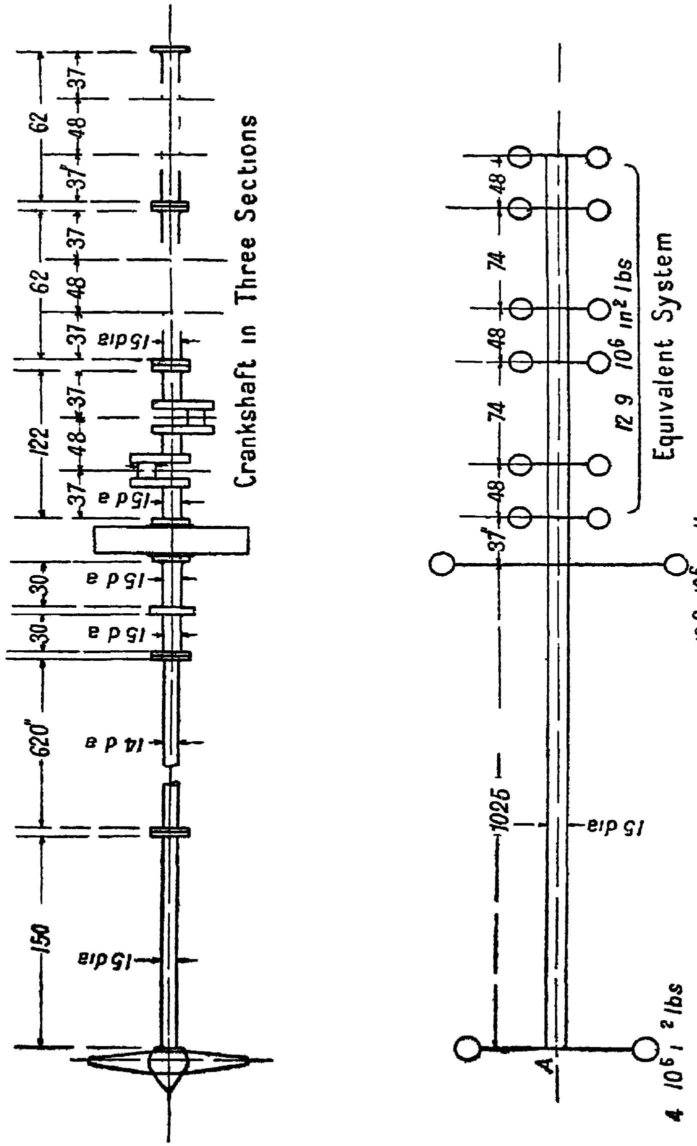


FIG 66

Fly wheel Effects —

$W\,K^2$ for propeller $=10\ 000 \times 20\ \ =4\ 000\ 000$ in lb

$W\,K^2$ for crank masses $=(4500+2500)\ 6 \times 17\ 5$

$=12\ 900\ 000$ in lb

$W\,K^2$ for fly wheel $=12\ 000 \times 40^2=19\ 200\ 000$ in ^{2} lb

Position of Node —Take moments of fly wheel effects about A —

$$19\ 20 \times 1025=19\ 700$$

$$12\ 9\ (1025+37+48+74+24)=15\ 600$$

$$\text{Total}\quad 35\ 300$$

The distance of the node from $A=\dfrac{35\ 300}{36\ 1}=978$ in

Dealing with the part of the system to the left of the node and applying equation (1) —

$$F=9\ 55\sqrt{\frac{4960 \times 12 \times 10^6 \times 386}{978 \times 4 \times 10^6}}=730 \text{ oscillations per minute}$$

And the critical speed $\dfrac{730}{6}=121\ 6$ R P M

which is practically the working speed of the engine

Critical speeds of the second and third order and so on are possible at 61 30 5 R P M etc but these are not as a rule important

It is sometimes useful to repeat the calculation for different weights of fly wheel and plot a curve connecting moment of inertia of fly wheel and critical speed so that the possible variation of critical speed obtainable by altering the fly wheel can be seen at a glance

Torsional oscillations about two or more nodes are also possible but as these involve higher speeds and are therefore more subject to damping it is doubtful if they are of much practical importance

To find the Moment of Inertia of a Fly wheel —In the first instance suppose the wheel in question is a disc wheel i e a solid of revolution Referring to Fig 67 the thick full line represents the section of the wheel Z Z is the axis and S S is a line through the extreme radius of the wheel parallel to the axis at a distance R from the latter Rule any line A B parallel to the axis cutting the outline of the section in A and B Project A and B on to S S at C and D Join C and D to any

convenient point O on the axis cutting A B in A_1 and B_1 Proceed similarly with different positions of the line A B and join up the various positions of A_1 and B_1 thus obtaining a new figure—the First Derived Figure Treat this figure as though it were the original figure and obtain the Second Derived Figure Similarly with this figure obtaining the Third Derived Figure

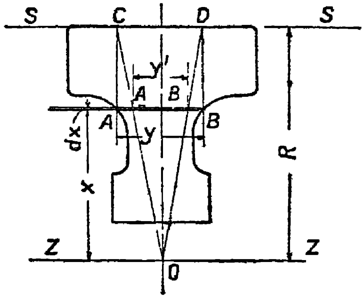


FIG 67

Let A = Area of original section in sq in

A_1 = Area of First Derived Figure in sq in

A_2 = Area of Second Derived Figure in sq in

A_3 = Area of Third Derived Figure in sq in

w = Weight in lb of one cubic inch of the material

$$\text{Then } A=\int_0^R y\,dx \quad A_1=\frac{1}{R}\int_0^R x\,y\,dx \quad A_2=\frac{1}{R}\int_0^R x^2\,y\,dx$$

$$A_3=\frac{1}{R_3}\int^R x^3\,y\,dx$$

$$\text{Weight of wheel}=2\pi\,w\int_0^R x\,y\,dx=2\pi\,w\,R\,A_1$$

$$\text{Moment of inertia of wheel}=2\pi w\int_0^R x^3 y\,dx=2\pi w\,R^3 A_3$$

$$\text{Radius of gyration}^2=P^2\frac{A_3}{A_1}$$

The above hold good for any position of the line S S which may therefore be taken where most convenient In cases where the section tapers towards the extreme radius (a screw propeller for instance) the line S S is best located at a distance of about one half or one third of the extreme radius from the axis

In the case of a screw propeller or a fly wheel with arms the rotating body must first be reduced to an equivalent disc wheel This is readily done as follows Describe a radius R which cuts through the arms or blades as the case may be Divide the total area of section at this radius by 2π R and the result is the thickness of the equivalent disc at this radius Repeat for a number of different radii covering the whole range

Types of Fly wheels —Fig 68 shews a disc wheel cast in one

piece and provided with a number of drilled holes in the rim for turning the engine by means of a bar Degree marks are cut on the edge of the rim to facilitate valve setting Fig 69 shews a disc wheel with a separate centre an arrangement which

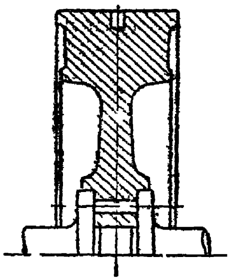


Fig 68

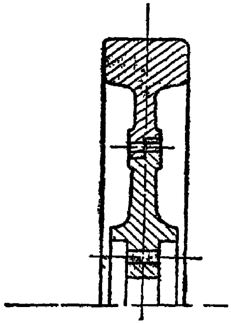


Fig 69

makes it easier to obtain a sound casting Large wheels are usually cast in two pieces and Fig 70 shews a design which is suitable for weights up to at least 20 tons It should be noted that no keys are provided for securing the wheel to the shaft If the boss of the wheel is bored one thousandth per inch of

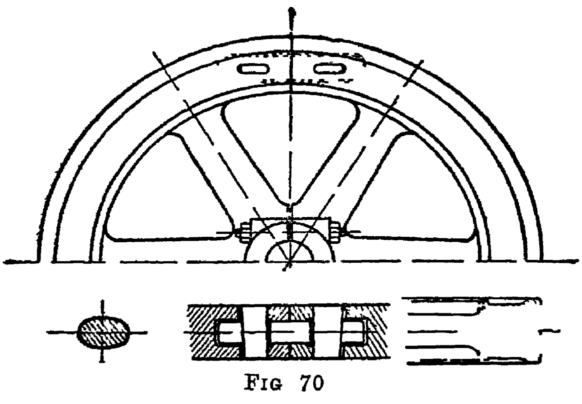


Fig 70

diameter less than the shaft and the bolts are drawn up at about the temperature of boiling water the frictional grip is quite sufficient for the largest wheels and the danger of splitting the boss of the wheel involved in the use of keys is avoided The same applies to pulleys for belt or rope drives A rather more elaborate wheel in which greater precautions have been

taken is shewn in Fig 71 In this case it is advisable to make the bore of the wheel the same as the shaft diameter and to give a shrinking allowance of about one thousandth per inch of diameter to the bore of the shrunk ring For large stationary engines some form of barring gear is necessary and where electric power is available a motor driven gear is a great convenience For marine engines a worm or other self locking gear is essential and where the auxiliaries are electrically driven an electric turning gear should be fitted as the use of the latter greatly expedites adjustments to the valve gear

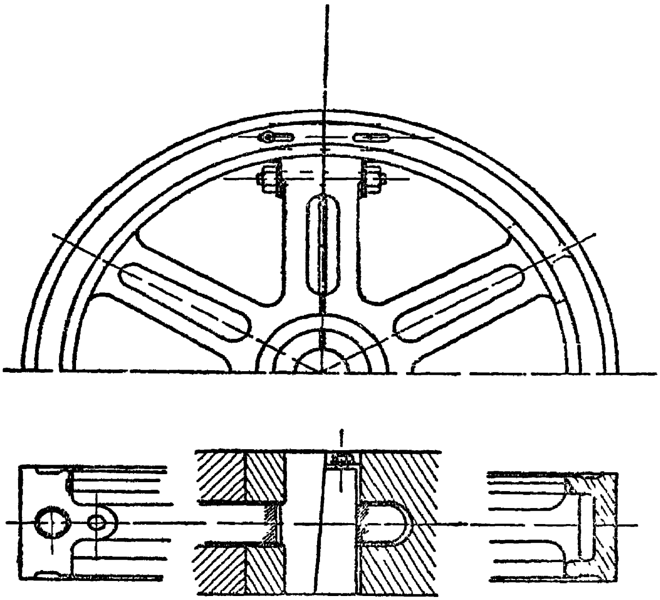


FIG 71

Strength of Fly wheels —Continental practice favours a peripheral speed of about 100 feet per second for cast iron fly wheels and this corresponds to a stress of about 1000 lb per sq in An investigation by Mr P H Smith into the case of a split fly wheel which burst at Maidenhead in 1912 shewed that the engine (the fly wheel of which had a normal working peripheral speed of about 100 feet per second) was running about double its normal speed and as the stress varies as the square of the speed it follows that the factor of safety under normal conditions was about 4 Destruction tests of wheels and models of wheels shew that the bursting speed is about 200 feet

per second for split wheels and 400 feet per second for solid wheels The discrepancy seems very large and difficult to account for Average British practice is in favour of a slightly lower peripheral speed (about 90 feet per second) With marine engines considerations of space generally necessitate a still lower figure The strength calculations for a fly wheel will be illustrated by an example

Example Required to find the approximate dimensions of a fly wheel suitable for 180 revolutions per minute given that

$$W K^2 = 40\ 000\ 000 \text{ in}^2 \text{ lb}$$

Peripheral speed 100 ft per sec

Maximum twisting moment due to engine 450 000 in lb

Outside radius of wheel $= \dfrac{100 \times 12 \times 60}{2\pi \times 180} = 63\ 7$ in say 64 in

Take inside radius of rim $= 52$ in

Then radius of C G of rim section $= 58$ in

Let B = width of rim Then —

$$\text{Weight of rim} = 12 \times B \times 2\pi \times 58 \times 0\ 26$$

And approximate moment of inertia

$$= 12 \times B \times 1\ 64 \times 58^3 = 40\ 000\ 000 \text{ in}^2/\text{lb}$$

From which $B = 10\ 5$ in

Since the stress due to a peripheral speed of 100 ft per sec is about 1000 lbs /in ^{2} the total ension at each joint of the rim is equal to $12 \times 10\ 5 \times 1000 = 126\ 000$ lb for which pull the dowel and cotter section must be designed

Allowable stress in dowel say 6000 lb per sq in

Effective area of dowel section $\dfrac{126\ 000}{6000} = 21$ sq in

Since about one third of the section of the dowel is cut away by the cotter hole (see Fig 72) the gross sectional area of the dowel must be $\dfrac{21 \times 3}{2} = 31\ 5$ sq in say 4 in $\times$ 8 in

Thickness of cotter $= \dfrac{8}{3} =$ about $2\frac{3}{4}$ in

Bearing pressure of cotter on dowel

$$\frac{126\ 000}{2\ 75 \times 4} = 11\ 450 \text{ lb per sq in}$$

which is allowable

If the hole for the dowel is made $\frac{1}{2}$ in wider than the dowel itself the bearing length for the cotter on the rim will be $10\ 5 - 4\ 5 = 6$ in and bearing pressure of cotter on rim

$$\frac{126\ 000}{2\ 75 \times 6} = 7650 \text{ lb per sq in}$$ also allowable

Allowable shear stress for cotter (which is in double shear) say 5000 lb per sq in

Depth of cotter $\frac{126\ 000}{5000 \times 2\ 75} = 9\ 2$ in say 9^{1} in over the rounded ends

The distance l between the inside edge of the cotter hole and the rim joint must be sufficient to obviate risk of the

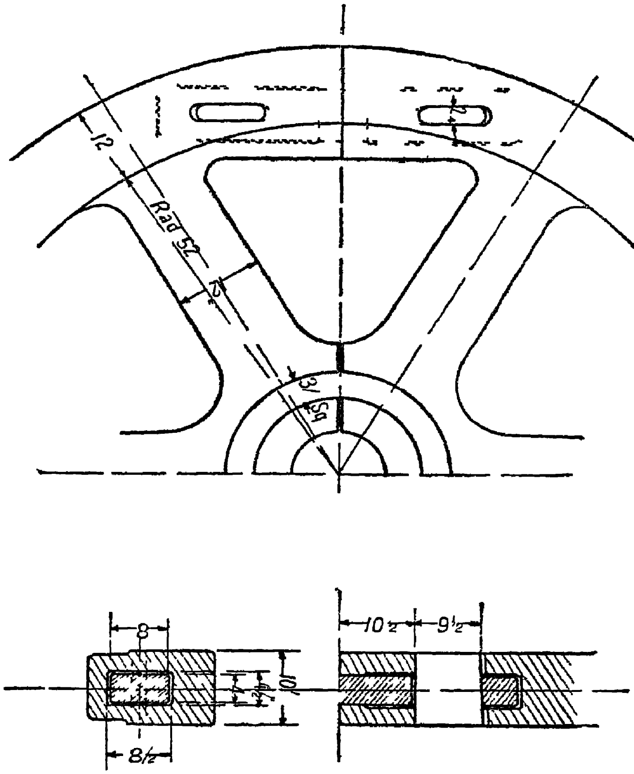


FIG 72

intervening metal being torn out in double shear (This point is sometimes overlooked in otherwise well proportioned rim joints)

Allowing a shear stress of 1000 lb per sq in —

$$l = \frac{126\ 000}{1000 \times 2 \times 6} = 10\ 5 \text{ in}$$

Owing to the difficulty of analysing the straining actions on the arms it is well to give the latter ample proportions

An approximate method of calculation is given below

Assume that the maximum twisting moment due to the engine (450 000 in lb) is transmitted to the rim by means of a constant shear force across the arms and that the bending moment is a maximum at each end of an arm and zero at the centre

The length of each arm from boss to rims is about 40 in and the distance of its centre from the centre of the wheel about 32 in

Then shear force in each arm $=\frac{450\,000}{6\times 32}=2340$ lb (assuming six arms)

And maximum bending moment at end of each arm $=2340\times 20=46\,800$ in lb

Taking a low stress of 500 per sq in to allow for direct tension in the arms bending modulus of arm section

$$=\frac{46\,800}{500}=93 \text{ in }^3$$

This is satisfied by a rectangle section 6 in $\times$10 in which could be replaced by an oval section about 7 in $\times$12 in to reduce wind resistance The bolts at the hub of the wheel are sometimes made as strong as the rim joint in which case the core area of two bolts will be the same as the net effective area of one dowel viz 21 sq in

This gives a bolt of about 4 in diameter If shrunk rings are employed these will have a square section about $10\ 5=(3\frac{1}{4}\text{ in })^2$

The above calculations must be regarded as preliminary only and give the draughtsman a basis on which to start designing The next step will be to check over the weight and radius of gyration of the complete wheel in the manner already described The dimensions and stress calculations will then be amended accordingly

Literature —For information on the strength of fly wheels see —Unwin W C and Mellanby A L The Elements of Machine Design Part II

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CHAPTER VII

FRAMEWORK

A LARGE number of different types of framework have been employed in Diesel Engine construction and a complete classification will not be attempted here The outstanding types in successful practice may however be broadly divided into a few well defined classes as under —

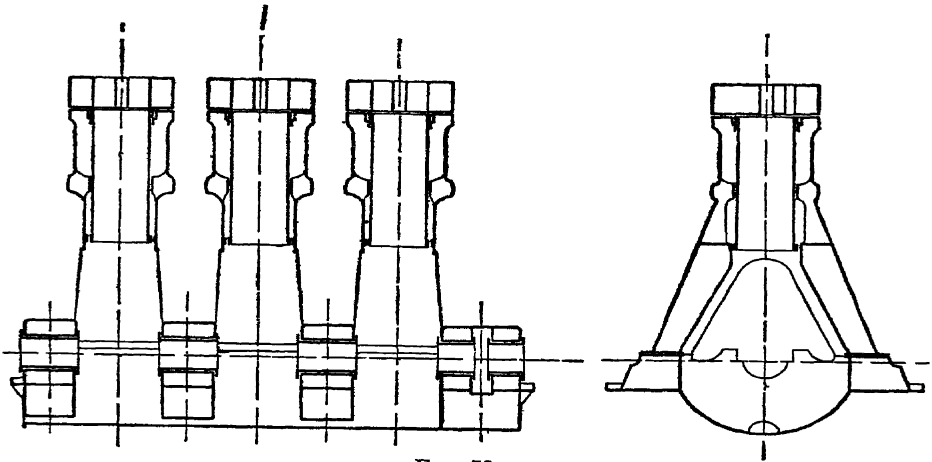


FIG 73

A **Frame Type** —This is the earliest type of Diesel Engine construction and on account of its merits is still very extensively used Referring to the diagrammatic drawing Fig 73 it will be seen that a stiff bedplate of box section is provided and that each cylinder stands on its own legs without support from its neighbours The legs of the column are cast integrally with the cylinder jacket into which a liner is fitted The breech end of the cylinder is closed by means of a deep cylinder cover of box section

The main tensile load due to the cylinder pressure is transmitted from the cover through the jacket and legs to the bedplate The reaction corresponding to this load occurs of course at the main bearings and consequently that part of the bedplate between the column feet and the main bearing housings must be designed to deal with the bending moment occasioned by the fact that the tensile load in the columns and the reaction at the bearings are not in the same plane Casting the cylinder jacket and column in one piece reduces fitting and machining operations to a minimum and the independence of the individual cylinders would appear to have no disadvantages so far as land engines are concerned

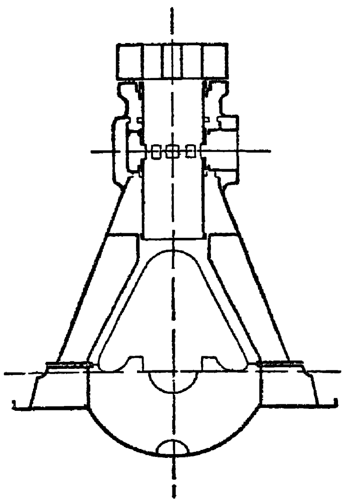


Fig 74

Fig 74 shews the same type of construction applied to a two stroke land engine and Fig 77 to a four stroke marine cylinder

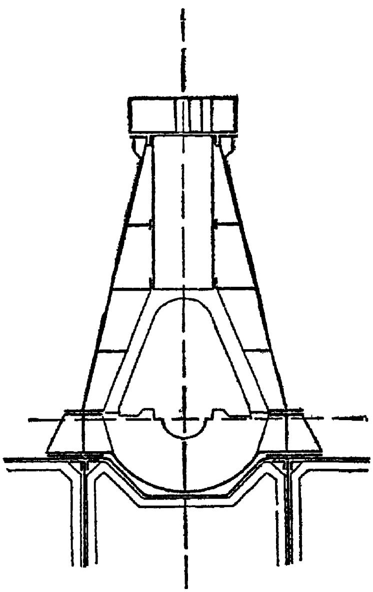


Fig 77

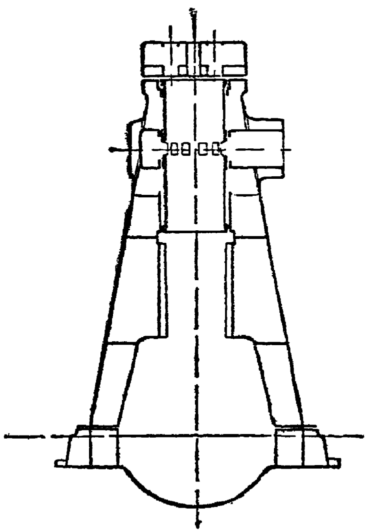


Fig 78

Slightly lengthening the column legs enables crosshead and guides to be fitted (see Fig 78 which represents a large two cycle land engine) Occasionally one of the column legs takes the form of a steel tie rod with a view to giving greater accessibility to the running gear and to enable the crank shaft to be replaced if necessary without dismantling the whole engine Unfortunately this arrangement nullifies many advantages of the A frame construction as special splash guards must now be fitted to retain the lubricating oil which office they do not always perform very efficiently and also additional

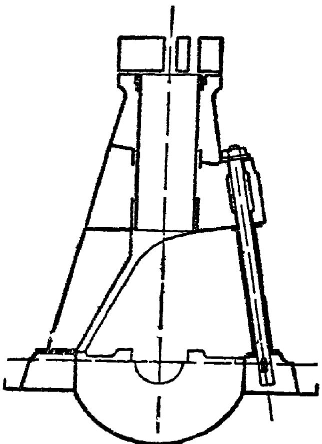


Fig 75

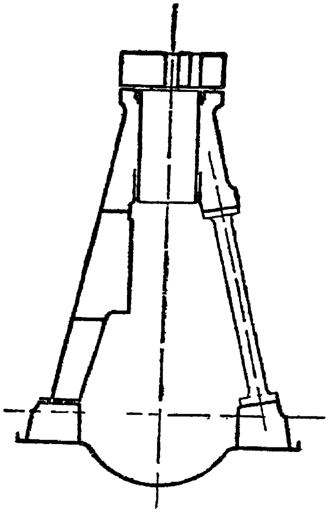


Fig 76

machining and fitting operations are introduced which add to the cost of production without increasing the efficiency of working Figs 75 and 76 shew this construction applied to trunk and crosshead engines respectively

Crank case Type —The crank case type of Diesel Engine was introduced when a desire was felt for higher speeds necessitating forced lubrication The crank case bears external resemblance to that of a high speed steam engine (see Fig 79) On the other hand the high pressures dealt with in the cylinder of a Diesel Engine necessitate the crank case being strengthened internally to an extent which is not found necessary in steam practice Sometimes the box or girder construction of the crank case is relied upon to transmit the tensile stresses from

the cover to the bedplate more frequently however steel staybolts are provided for this purpose (see Fig 80) The latter procedure however does not justify flimsy construction

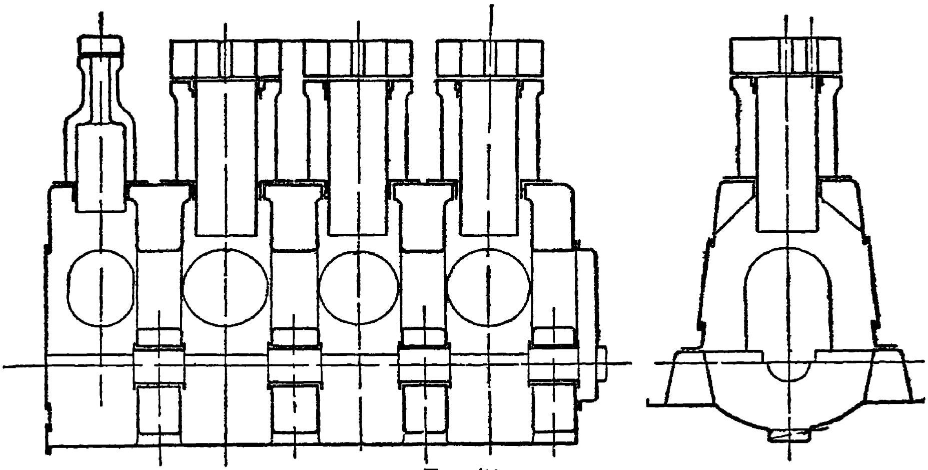


FIG 79

or a careless distribution of metal in the crank case as the guide pressure has still to be reckoned with and the pressure caused by tightening up the staybolts may be relied upon to cause serious distortion of a poorly ribbed case

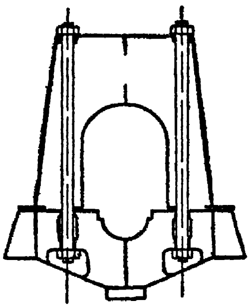


FIG 80

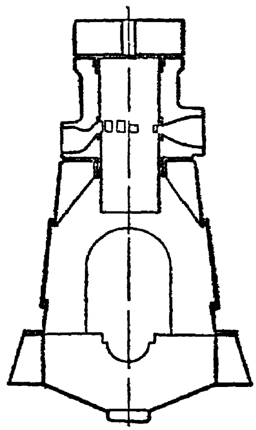


FIG 81

The cylinders being separate are secured either by a round studded flange or by passing the staybolts through each corner of a deep square flange of hollow section cast at the lower end

of the cylinder jacket for this purpose The latter arrangement requires four staybolts for each cylinder whereas with the former it is usual to arrange a pair of staybolts only at each main bearing girder With the crank case construction it is not necessary to make the side girders of the bedplate so strong as for an A frame type of engine as the bending action referred to above is avoided and the bedplate and crank case when bolted together form a girder construction of great rigidity On the other hand the upper part of the crank case is clearly subject to bending actions similar to those which

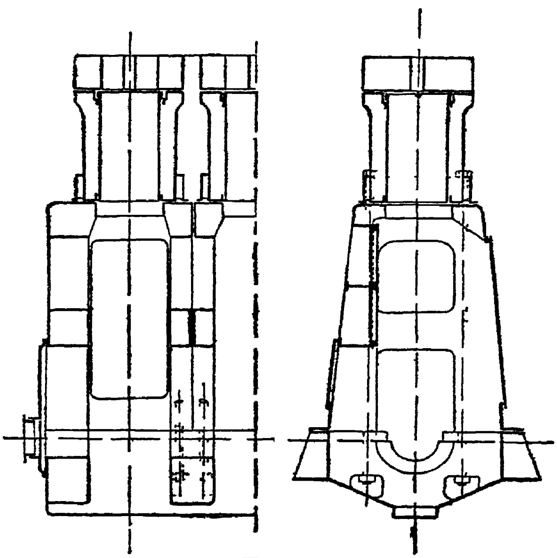


FIG 82

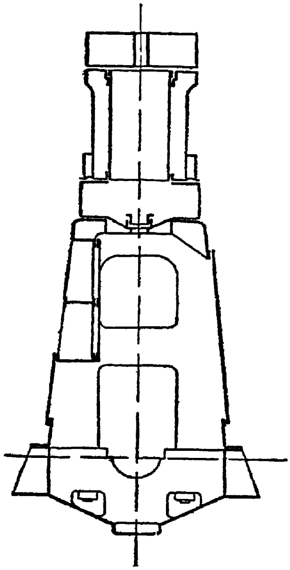


FIG 83

occur in the bedplate of an A frame engine and must be designed with this fact in view Fig 81 shews a section through a two stroke trunk engine of the crank case type Figs 82 and 83 shew the crank case construction applied to crosshead engines The suitability of this type of framework for marine service has been amply proved in practice In some cases the crank case is common to two or more cylinders and in others the case for each cylinder is a separate casting the individual cases being bolted together to form a virtually continuous box of great strength and rigidity

Trestle Type —With this construction illustrated in Fig 84

the cylinders are bolted to a base plate or entablature resting on trestle shaped columns the feet of the latter being secured to the main bearing girders If the guide casting itself and its attachment to the trestles are sufficiently strong this construction can compete with the crank case type of frame in the matter of rigidity The same effect could doubtless be achieved by some form of bracing In Fig 84 an alternative form of crosshead and guide is shewn to which the trestle arrangement

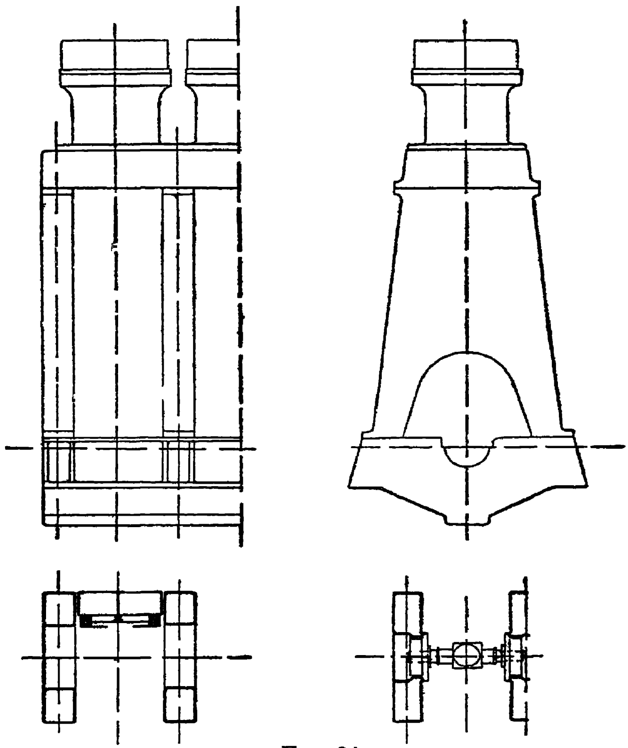


Fig 84

particularly lends itself viz the fore and aft double guide usually found on paddle steamers The advantages of this form of guide for Diesel Engines are —

1 Accessibility of running gear the piston cooling gear in particular
2 The guide blocks being free to adjust themselves to the guides are capable of sustaining a greater specific pressure and consequently require less bearing surface than the usual type of guide shoe

It is to be noted that the trestle construction involves some little extra care to retain lubricating oil when forced lubrication is used

Staybolt Construction —With this construction the cylinders are connected to the bedplate by means of turned bolts only and the saving in weight on this account amounts to the considerable figure of about 25% of the complete weight of the engine The reduction in cost of manufacture must also be considerable when the staybars are made of ordinary bright shafting screwed at the ends Examples are shewn in Figs 85

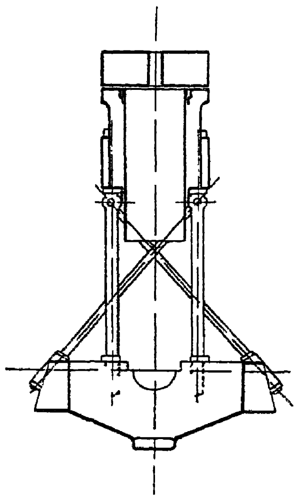


FIG 85

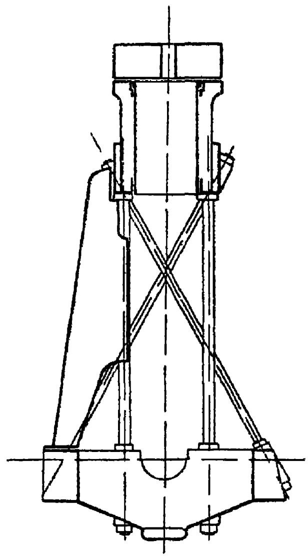


FIG 86

and 86 for trunk and crosshead engines respectively In the latter the cast columns are relatively light being designed for the guide pressure only and their top ends are made free to slide in order to avoid the subjection of the column to tensile strains on the compression and firing strokes When forced lubrication is used with such a construction special care is necessary in the detailed design of the casings for retaining oil These are either of well stiffened sheet steel supported on an angle iron framework or a combination of steel doors with light cast bracing pieces

Design of Bedplates —The design of a suitable bedplate

involves consideration of the following points which will be dealt with in order viz —

(1) The provision of a suitable main bearing
(2) A girder construction under each main bearing capable of supporting the full bearing load without central support
(3) A sufficiently strong and stiff connection between the main bearing girders forming at the same time an oil tight tray
(4) Suitable studding or staybolt arrangements for carrying the tensile pull of the columns
(5) Arrangements for supporting the cam shaft driving gear
(6) Means for collecting drainage of lubricating oil to some convenient sump whence it can readily be drawn off with a view to filtration and repeated use
(7) Facings for barring gear auxiliary pumps etc

Main Bearings —Examination of badly worn crank shafts indicates that the high bearing pressure obtaining for a short time when the piston is at its firing centre gives rise to far less abrasive action on the bearings than the less intense but longer sustained pressures due to inertia and centrifugal force in a four cycle engine It appears that a film of oil is capable of sustaining a heavy pressure for a short time but once the film has broken down relatively feeble pressure is sufficient to cause abrasion and there is small chance of the surfaces receiving a new film of lubricant until the pressure is removed The result is that very little trouble is experienced with the lubrication of the main bearings of four stroke engines (the pressure on the journals being frequently reversed) unless the peripheral speed is so high as to reduce seriously the viscosity of the oil film by means of the heat generated by friction

With the highest speeds at present used in practice this contingency seems to have been successfully avoided by the use of forced lubrication without resorting to water cooled bearings

With two cycle engines the direction of pressure is probably not reversed at all in most cases and lubrication is consequently more difficult When the peripheral speed is low and the oil film in consequence as stable as possible satisfactory results are obtainable even with ring lubrication of good design if the maximum bearing pressure is kept about

30% lower than would be considered good ordinary practice with four stroke engines If on the other hand high peripheral speeds or moderate bearing surfaces or both are required then a system of pressure forced lubrication would appear to be necessary preferably in conjunction with a system of water cooling in extreme cases The following table gives a rough guide to the limitations of the various systems of main bearing lubrication —

System of Lubrication	Peripheral Speed of Journal feet per minute	Projected Area of One Journal expressed as percentage of the Area of Piston
FOUR STROKE ENGINES—		
Ring lubrication	550	55%
Forced lubrication	750	40%
Forced lubrication and water cooling	above 750	40%
TWO STROKE ENGINES—		
Ring lubrication	550	75%
Forced lubrication	700	60%
Forced lubrication and water cooling	above 700	60%

Ring lubrication for main bearings at one time the usual practice for land engines is now almost confined to horizontal engines which do not apparently lend themselves to the usual system of forced lubrication

Ring Lubricated Main Bearings —These are similar to the bearings fitted to electrical machinery and need not be described in detail The arrangements for catching the oil squeezed out of the bearings and conveying it back to the oil well merit careful attention as inefficiency in this direction leads to unnecessary waste of oil In particular the oil spaces and holes should be as large as possible to avoid congestion Fig 87 shews a very usual form

Forced Lubricated Bearings —These follow high speed steam engine practice very closely and the usual form of

staggering the circumferential oil groove in the top and bottom brasses is commonly adopted See Fig 88

Main Bearings Shells —These are of cast iron or steel in commercial work and in the best practice the shells are tinned previous to the white metal being poured in The chief essentials are —

(1) Adequate thickness of shell
(2) Good quality of white metal
(3) Good adhesion between white metal and shell

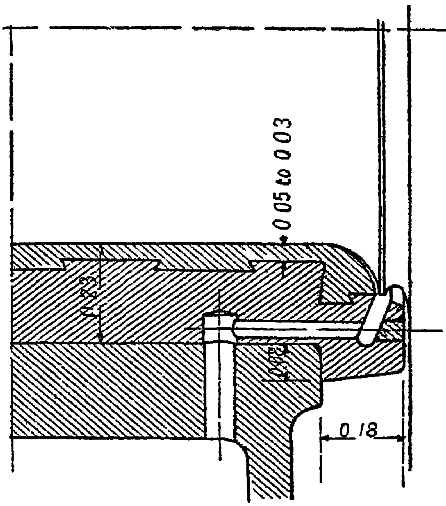


FIG 87

Common proportions are shewn in Figs 87 and 88 the unit being the diameter of the journal

The bearing cap should be designed as a beam capable of carrying a central load equivalent to the full inertia and centrifugal load due to one set of running gear This is possibly a

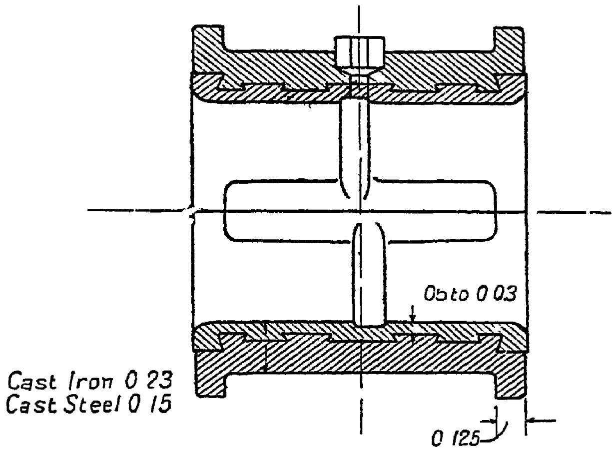


FIG 88

little on the safe side but reference to Chapter V will shew that the margin is not large in the case investigated there

Main Bearing Girder —Where forced lubrication is used

the main bearing girder may conveniently be of I section the bottom flange being formed by the oil tray for ring lubrication a box section lends itself more conveniently to the formation of the oil reservoir In any case the box section is preferable in the larger sizes The depth of the girder is determined by that of the oil tray required to give an inch clearance or so to the connecting rod big end at the bottom of its path Referring to Chapter V it will be seen that the maximum reaction at a bearing for the case considered is equal to 0 8 of the resultant load due to pressure inertia and centrifugal force and this is the load for which the girder must be designed In other cases the load may be less than this but it is doubtful if in any case it approximates to the conventional load frequently assumed viz one half the resultant cylinder load

A very debatable point is the extent to which the oil tray can legitimately be regarded as a part of the tensile flange of the girder The author s practice in this respect is to ignore the middle half of that part of the tray lying between two bearing girders (see Fig 89) The span of the girder is the distance between the two points at which it meets the side girders

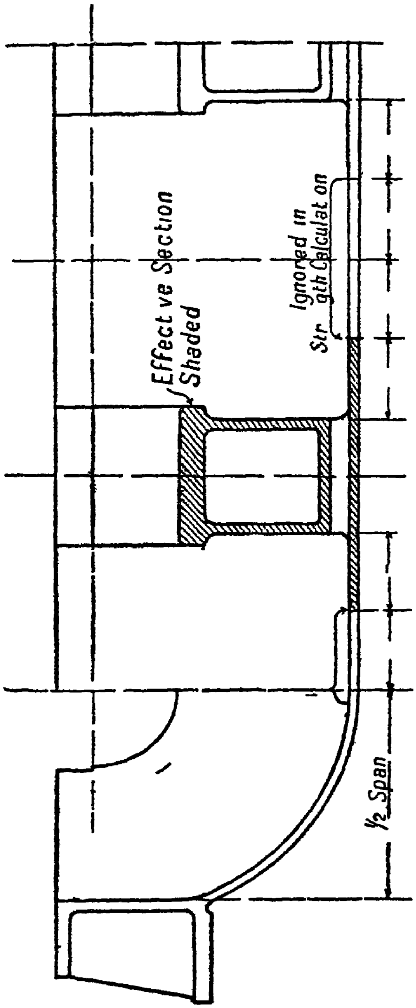


Fig 89

If W = Load on girder in lb
l = Span in in
M = Bending moment in in lb

Then $M = 0.2\ Wl$ — approximately

The assumption being that the fixing moments at the ends are negligible (which if not correct

is on the safe side) and that the load is distributed over the journal Allowable stress 1500–2500 lb per sq in for cast iron

Side Girders—With A frame engines the bending moment on each side girder may be taken as —

$$\frac{\text{Half pressure load} \times \text{Distance between centres of bearings}}{6}$$

The usual stress allowance being about 1500 lb per sq in Where the trestle or crank case type of frame is used the side girders may be of lighter section (proportions will be given later)

Arrangements for Carrying Tensile Pull of Columns—With the A frame construction the foot of each column is secured by a row of studs the stress in which when referred to the normal maximum working pressure of 500 lb per sq in in the cylinder amounts to about 5000 to 10 000 lb per sq in according to the size of the stud It is very convenient to have a list of the loads which studs and bolts of different sizes can conveniently carry and such a list is given below —

Size of Bolt or Stud (Whitworth)	Stress (Core) allowed lb / n	Working Load lbs
$\frac{1}{2}$	2000	240
$\frac{5}{8}$	2850	550
$\frac{3}{4}$	3550	1080
$\frac{7}{8}$	4250	1800
1	5000	2750
$1\frac{1}{8}$	5250	3650
$1\frac{1}{4}$	5500	5000
$1\frac{3}{8}$	6000	6300
$1\frac{1}{2}$	7100	9300
$1\frac{3}{4}$	8500	15 000
2		20 000
$2\frac{1}{4}$		24 000
$2\frac{1}{2}$		32 000
$2\frac{3}{4}$		37 000
3		46 000

Care must be taken that none of the studs are at any considerable distance from adequate supporting ribs This is best

obtained by judicious spacing of the studs rather than the provision of special ribs for the purpose

With the crank case and trestle types staybolts are usually fitted and in land work at any rate these should terminate within the bedplate and not penetrate to the under side of the latter for fear of oil leakage which would destroy the concrete The studs or bolts used to secure the crank case to the

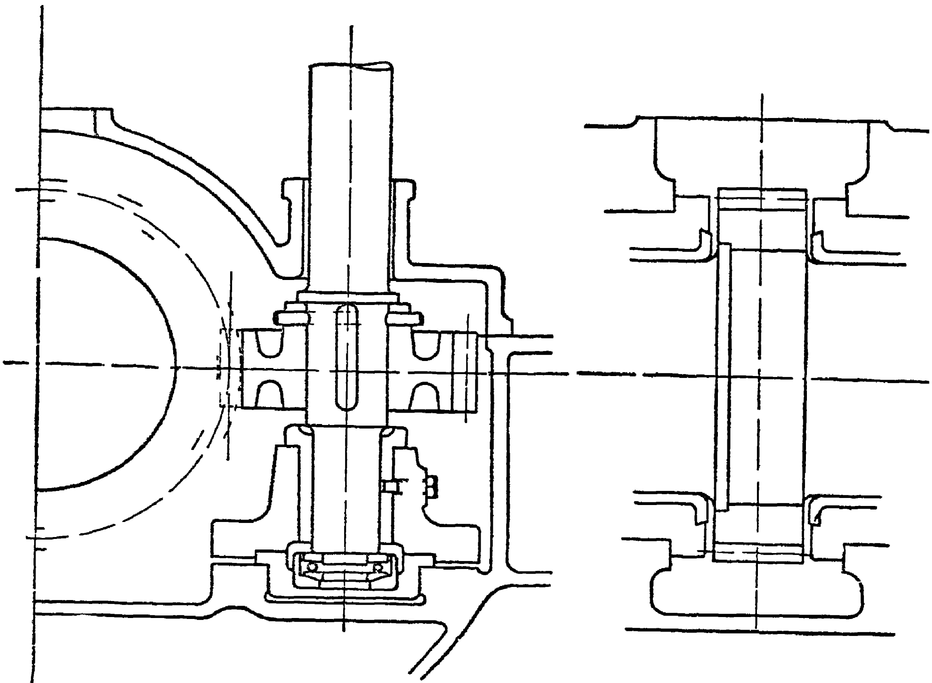


Fig 90

bedplate may be disposed more with a view to making an oil tight joint than to carry any definite load If staybolts are not fitted then a sufficiency of effective bolt or stud area must be arranged in the neighbourhood of each column foot and some of the bolts or studs must be inside the crank case

Cam shaft Driving Gear —The motion required by the valve gear is derived from the crank shaft by spiral or spur gearing in the majority of designs Fig 90 shews a very common arrangement of spiral drive with the driving wheel between the two sections of a divided main bearing It is good

practice to make the combined length of the two sections about 50% greater than the length of a normal bearing There would appear to be nothing against having the spiral wheel outside the bearing altogether provided the gear is at the fly wheel end This position for the valve gear drive is preferable to the compressor end as the weight of the fly wheel tends to keep the journal in contact with its lower bearing shell whereas the forward journal has freedom of motion (under the influence of forces which vary in direction) to the extent of the running clearance

In six cylinder engines the spiral gear is frequently arranged at the centre of the engine where it is very easily accommodated There seems to be some feeling that the cam shaft would whip unduly if driven from the end This difficulty (if any difficulty can be said to exist) is easily overcome by making the cam shaft about 10% larger in diameter than would be considered sufficient for a four cylinder engine

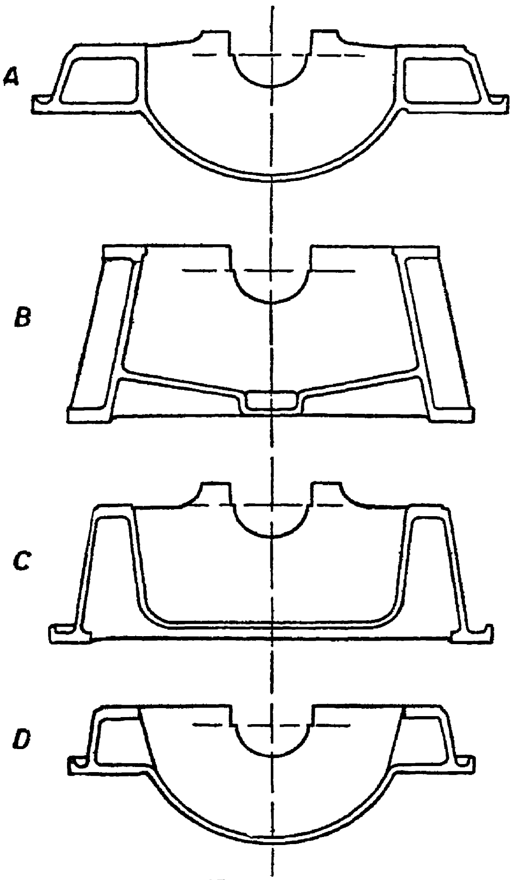


FIG 91

Where spur gearing is used for the valve gear drive facings must be provided for the support of the first motion shaft

Oil Drainage — With land engines of the non forced lubricated type the oil which drips down from the cylinders and is thrown from the big ends is drained periodically from the forward end of the bedplate and holes are cored through the main bearings girders to give the oil free passage Perhaps the best arrangement is a rectangular duct about four inches square running down the centre of

the oil tray Small holes are useless as they are easily choked With forced lubricated engines the same arrangements are made with the addition of a collecting sump of good capacity a pump for forcing the oil into the bearings and filters in duplicate These features being familiar in steam engine practice need not be described in detail It must be borne in mind however that where trunk engines are being considered the oil is contaminated with carbon so that the filtering arrange ments require to be on a more liberal scale than is necessary with engines in which the cylinder is isolated from the crank case

Proportions of Bedplate Sections —Fig 91 gives approxi mate proportions for various types of bedplate sections the unit being the stroke of the engine Type A is usually associated with the A frame construction Type B is a useful one for main or auxiliary marine engines as it enables the engine to be bolted direct to a tank top or to a deck with out building up a special seating

Type C is preferable to type A for land generating sets as the extra depth of bedplate enables the generator to be flush with the engine room floor without the necessity of building the engine on an unsightly plinth A deep bedplate is also very desirable with six cylinder engines as the cancella tion at the centre of the engine of the inertia and centrifugal couples gives rise to vibrations the amplitudes of which are reduced by increasing the stiffness of the framework

The general thickness of metal may be about 4% of the cylinder bore increased to about 6% or 8% on machined surfaces These figures are usually exceeded on small engines on account of the difficulty of obtaining consistent results in the foundry with thin metal The following figures for different diameters of cylinder represent good practice —

Bore of Cylinder in	General thickness of Metal for Bedplate in
10	$\frac{5}{8}$
12	$\frac{3}{4}$
15	$\frac{13}{16}$
18	$\frac{15}{16}$
21	$1\frac{1}{16}$
24	$1\frac{3}{16}$
27	$1\frac{1}{4}$
30	$1\frac{3}{8}$

The above are useful as a guide but must not be relied upon without check calculations as special constructions may require local strengthening to keep the stresses within a safe figure

A Frames —An A frame for a four stroke trunk engine is shewn in Fig 92 the discussion of those parts which are common to separate cylinders as distinguished from cylinders combined with columns will be reserved for a separate chapter The liner is a push fit or light shrink fit in the upper flange of the jacket The fit at M should be a few thousandths slack to prevent seizure at this point The jacket is swelled locally to give adequate water passage and six or eight strong ribs are provided to transmit the tensile load across this section which would otherwise be greatly weakened by the discontinuity of the wall At Q the liner is a push fit allowing the liner to expand axially relatively to the jacket Sometimes a stuffing box is fitted to prevent water leakage P is the water inlet connection L L are bosses to accommodate lubricator fittings N is a cleaning door

Strength of A Frames —Fig 92 is drawn for a 15 in cylinder the stroke being 21 in and the dimensions given re present good average practice The stresses may be checked as follows —

Maximum working pressure 500 lb per sq in

load $0\ 784 \times 15 \times 500$

$= 88\ 000$ lb

Tensile stress in parallel part of jacket (mean dia $= 23\ 5$)

$$= \frac{88\ 000}{\pi \times 23\ 5 \times 1} = 1190 \text{ lb /in}^2$$

Next consider section AA of one leg For this section conditions are worst if the nuts at the foot are not tight and the reaction at the foot consists of a simple vertical pull of 44 000 lb Referring to Fig 92 the direct tensile pull in the leg is 42 000 lb and the sectional area at AA being 57 sq inches

Direct tensile stress at AA	$= 42\ 000 - 57 = 737$ lb per sq in
Shear stress	$= 13\ 500 - 57 = 237$
Bending moment	$= 44\ 000 \times 9\ 5 = 42\ 000$ in lb
Section modulus	$= \sim 230$ in^3
Bending stress	$= 420\ 000 - 230 = 1825$ lb per sq in
Total tensile stress	$= 1825 + 737 = 2562$

This stress refers to the outside fibres of the column and is probably in excess of what actually occurs as the fixation of the foot by the holding down studs produces a moment in the reverse direction

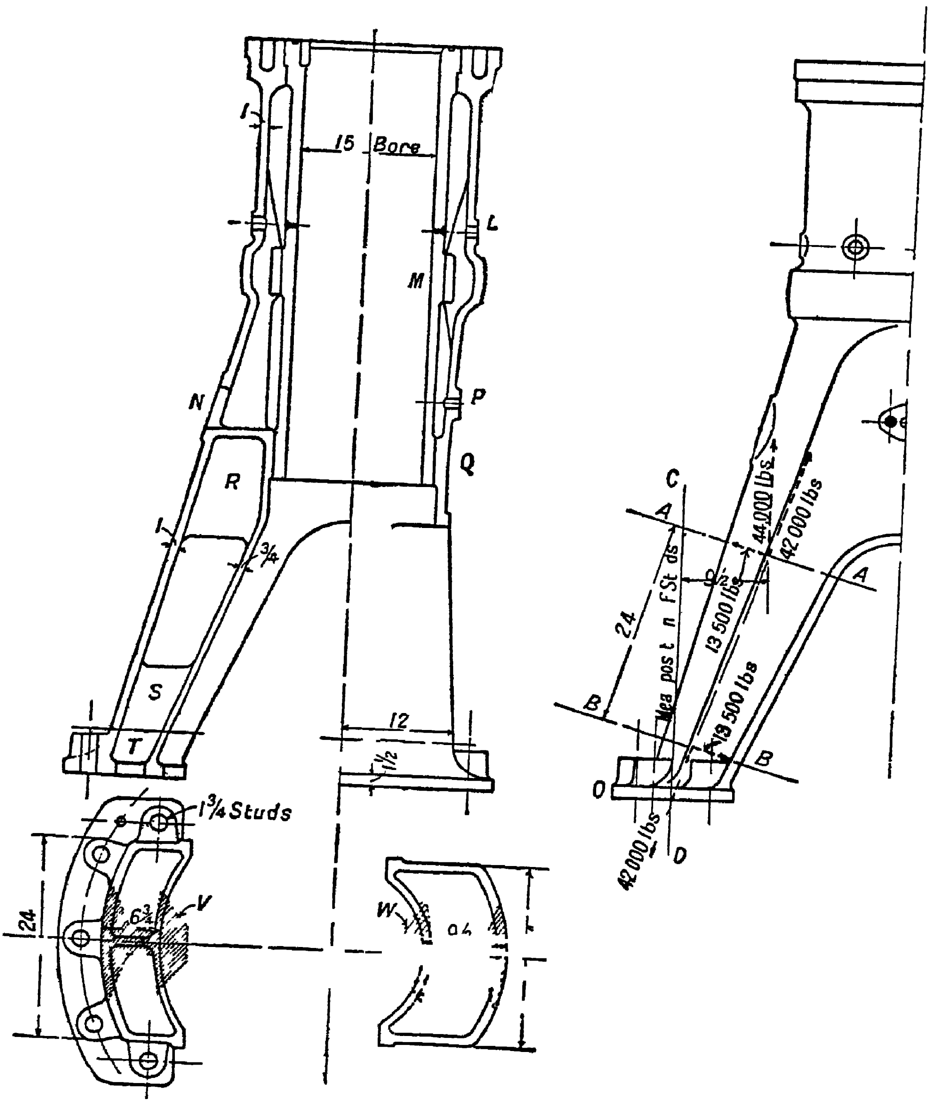


FIG 92

Section BB is subject to maximum stressing action if the foot is securely fixed to the bedplate as it should be The vertical reaction of 44 000 lb is again resolved into a direct pull along the axis of the leg amounting to 42 000 lb and a shear of 13 500 lb The area BB being 64 in ^{2} therefore the direct tensile stress at BB $= 42\,000 - 64 = 657$ lb /in ^{2}

The shear load of 13 500 lb at BB being opposed by an equal but opposite shear load at AA there must be a couple of magnitude $13\,500 \times 24$ in lb to keep the part of the leg lying between AA and BB in equilibrium Assuming that this couple is composed of two equal parts operating at AA and BB

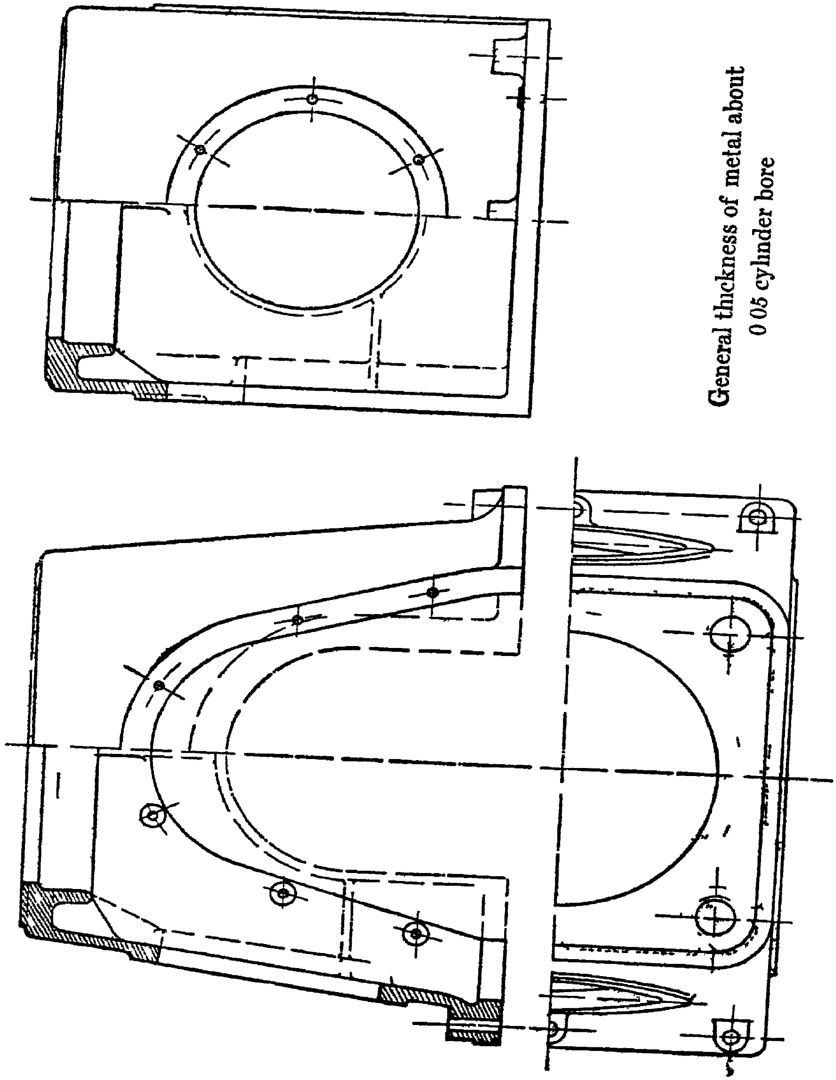
Bending moment at BB	$= 13\,500 \times 24 - 2 = 162\,000$ in lb
Section modulus	$= \sim 170$ in ^{3}
Bending stress	$= 162\,000 - 170 = 953$ lb per sq in
Total tensile stress	$= 953 + 657 = 1610$

The stress in the studs depends upon the degree to which the nuts are tightened and if of sufficient area the stress is probably not affected by the applied load It therefore only remains to see if the loads induced by tightening them up to their nominal working stress are sufficient to prevent the joint opening

Referring to the table on page 129 the nominal load of five $1\frac{3}{4}$ in studs $= 75\,000$ lb Subtracting 44 000 lb there remain 31 000 lb to resist tilting about the outside edge 0 of the foot The distance from 0 to the centre of mean position of the studs is $6\frac{1}{2}$ in and the corresponding moment is therefore $31\,000 \times 6\ 5$ $= 202\,000$ in lb which is greater than the moment to be resisted

Design of Crank cases —A simple form of crank case is shewn on Fig 93 stroke to bore ratio 1 25 The crank case is machined on each side and in position is bolted to its neighbours an arrangement which though unusual has its advantages both in the factory and outside as the small sections are easier to handle than a crank case in one piece Provision is made for staybolts and the thicknesses of metal shewn are about the minimum to give satisfactory results with this design It must not be thought that these proportional thicknesses are capable of substantial reduction with large sizes of engine without modification to the distribution of metal On the other hand if the interior of the case is built up in girder or box formation

or generally reinforced by internal ribbing as shewn dotted in Fig 93 the general thickness may be reduced by about 25% and advantage is taken of this fact in designing large engines the castings of which would be of undesirable thickness if the simpler forms were adopted In the event of staybolts not being used it is desirable to check the stresses in the legs in the manner described for an A frame Judging by successful



General thickness of metal about 0 05 cylinder bore

Fig 93

designs the use or omission of the staybolts has little influence on the strength of crank case required and this is readily explained by the fact that whereas the staybolts relieve the crank case of tensile stresses the tightening of the former throws a heavy buckling load on the crank case perhaps double the tensile load due to the working pressure in the cylinders These considerations do not apply however to those designs in which the staybolts are extended upwards to the cylinder cover In these cases the crank case has only the guide pressure to contend with On the other hand the use of

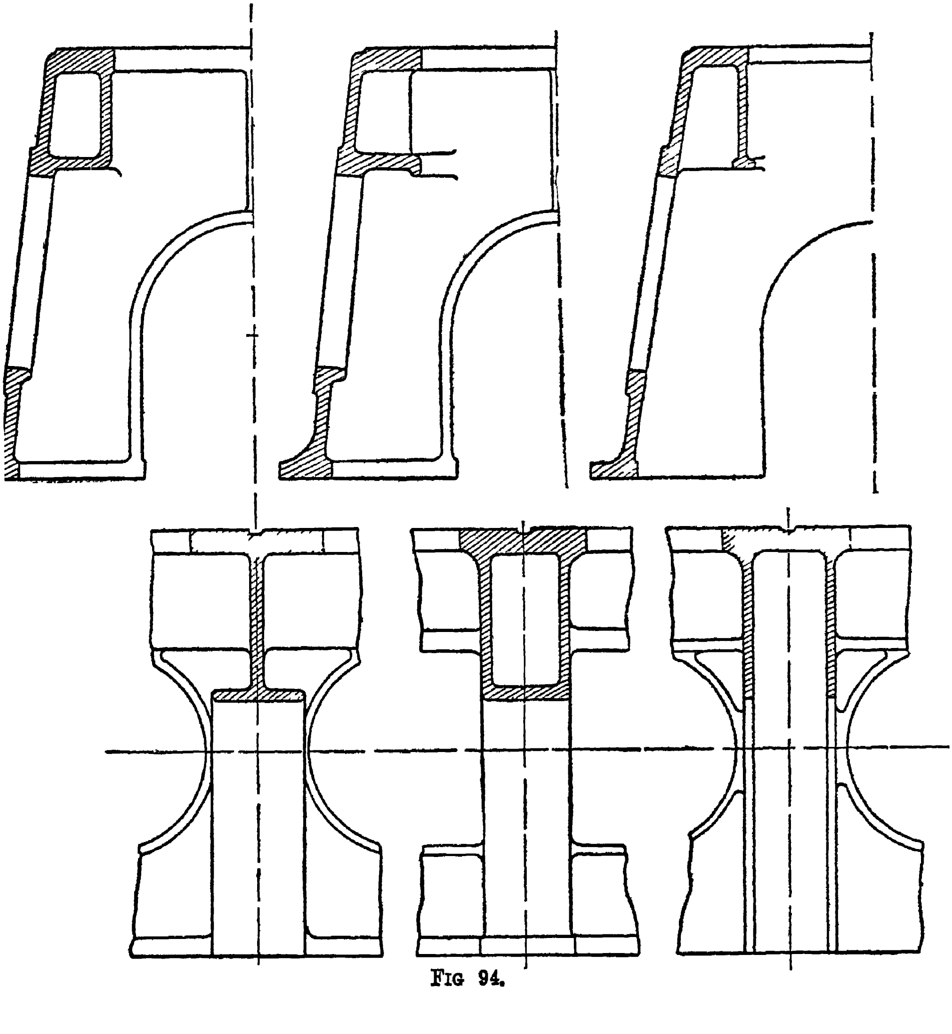


FIG 94.

staybolts passing through lugs on the cylinders enables the latter to be pitched closer together than would be easily possible otherwise and in any case they add strength and rigidity at very little expense and increase in weight

The above notes on the strength of crank cases as well as the figures for the thickness of metal apply equally to cross head as to trunk engines The additional height of the former has little if any influence on the matter as the guide reaction acts at the same height above the centre line of the crank shaft assuming the same length of connecting rod in each case Some alternative forms of construction are shewn in Fig 94 In the case of very small engines the use of the minimum thickness of metal allowable on considerations of strength and rigidity only would give rise to trouble in the average foundry and additional thickness is usually given In the following table foundry considerations are neglected as these must be dealt with by the designer in each individual case in accordance with his judgment of the capabilities of the foundry in question and in this matter the foundry manager will be able to give assistance

Bore of Cylinder in	General thickness of Crank case metal n Pla n type F g 93	General thickness of Crank case metal in Box or girder formation Fig 94	Diameter of Whitworth Staybolts in
10	$\frac{1}{2}$	$\frac{[illegible]}{16}$	$1\frac{[illegible]}{8}$
12	$\frac{5}{8}$	$\frac{1}{2}$	2
15	$\frac{3}{4}$	$\frac{[illegible]}{16}$	$2\frac{1}{[illegible]}$
18	$\frac{7}{8}$	$\frac{3}{4}$	3
21	$1\frac{[illegible]}{16}$"	$\frac{7}{8}$	$3\frac{1}{[illegible]}$
24	$1\frac{[illegible]}{16}$"	1	4
27	$1\frac{3}{8}$	$1\frac{1}{16}$	$4\frac{3}{8}$
30	$1\frac{1}{[illegible]}$	$1\frac{3}{16}$	$4\frac{3}{4}$

The figures for the diameters of the staybolts are based on the assumption that they carry the whole pressure load In cases where the cylinders are secured to the crank case by a studded flange the staybolts if fitted at all may be made considerably lighter according to judgment or the results of experiment Other points to be considered in designing a crank case are —

(1) The provision of oil tight access doors of ample size for overhauling the bottom ends
(2) End casings provided with oil flingers stuffing boxes or other means of preventing the escape of oil
(3) Facings and other necessary accommodation for valve gear
(4) Bosses to carry lubrication oil connections to the main bearings
(5) Facings for platform brackets
(6) A vent pipe or valve of large area to relieve pressure in the event of an explosion in the crank case without loss of lubricating oil during normal working
(7) Steady pins to each section of the case to fix correct location

Machining the Framework generally —In designing all parts of an engine the designer will keep in mind the capabilities and limitations of the manufacturing plant and the operatives This is especially necessary in the case of the framework on account of the relatively large size of the parts Where the most modern type of face milling plant is available the element of size offers no difficulties and bedplates of 60 feet in length may be faced in one operation Where planing must be resorted to the capacity of the machines must be studied in the early stages of the design Machined faces should be arranged in as few different planes as possible and ribs or flanges projecting beyond those planes are to be avoided as much for convenience in machining as for the sake of appearances The simpler forms of girder or box girder construction are to be preferred to those designs in which alternate perforation by lightening holes and reinforcement by ribbing mutually defeat each other s object The lightest strongest and cheapest forms are to be attained with a minimum of holes and ribs when cast iron is used Large steel castings however are preferably lightened out almost to the extent of lattice work in order to facilitate rapid stripping of the cores after solidification and to minimise initial stresses

Literature —The different types of framework used in Marine Diesel Engine construction and the forces acting on them are discussed in the following paper —Richardson J The Development of High Power Marine Diesel Engines Junior I E April 20th 1914

CHAPTER VIII

CYLINDERS AND COVERS

General Types —The great majority of Diesel Engines are provided with cylinder liner jacket and cover as separate pieces as in Fig 95 which refers to a four cycle trunk engine Different arrangements have however been used successfully and deserve mention With small engines simplification is achieved by casting the jacket and liner in one piece as in Fig 96 Remembering that the bulk of the jacket wall remains

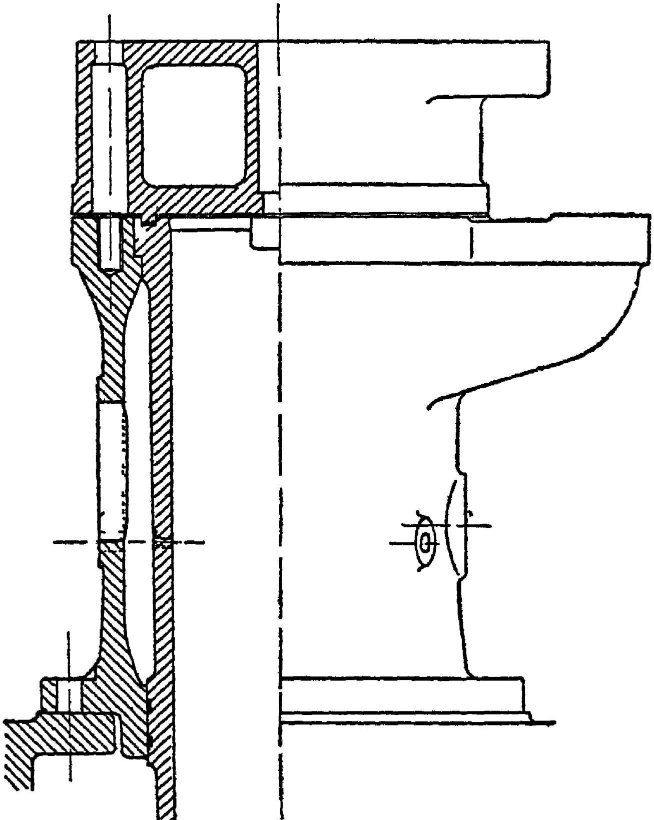


Fig 95

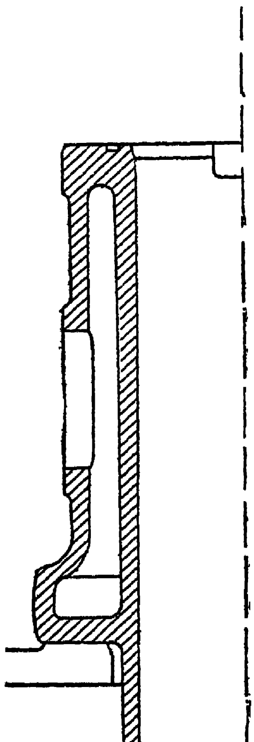


Fig 96

stone cold it will be appreciated that this construction involves increased tensile stresses on the jacket due to the tendency of the liner to expand and jackets of this kind have been known to crack circumferentially In cases where staybolts have been fitted to carry the tensile stresses from the cover downwards little damage has resulted On the other hand when the jacket has been relied on for this function rupture during working may easily occur and has sometimes resulted in the cylinder being projected towards the roof These considerations would appear to indicate that the use of this construction without staybolts or other safeguards is not lightly to be attempted without serious consideration of the capabilities of the foundry In Fig 97 is shewn a construction in which the cover is incorporated with the cylinder casting in motor car style In this case the tensile stresses are mainly carried by the liner and the jacket is made relatively thin and flexible This design though probably safer than that of Fig 96 also makes some demand on the skill and care of the foundry people In this connection it is worth while bearing in mind that many failures might possibly have been avoided if it had been realised that certain continental designs in which lightness has been a primary consideration were only practicable if the greatest care were exercised in the selection of material and in making the castings There are other types of cylinder in successful use notably those in which the liner and cover are cast together apart from the jacket but this chapter will be very largely devoted to the consideration of the details of the more common construction in which the liner jacket and cover are separate pieces Unless the contrary is stated cast iron is understood to be the material in each case

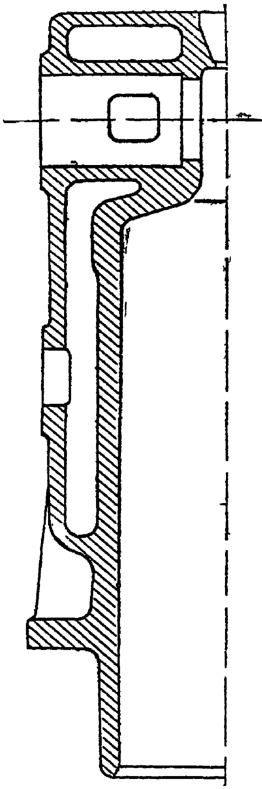


FIG 97

Liners —Special cast iron is used for liners but there is little unanimity of opinion as to the most desirable properties beyond the obvious requirements of soundness and homogeneity The grea est difficulty to be overcome is abrasion by the piston rings At present it seems open to question whether the

problem is most influenced by the material of the liner or the piston rings themselves Four stroke liners very seldom crack except on the occasion of the seisure of an uncooled piston This immunity is traceable to the very moderate heat flux to which four stroke cylinders are generally subject Two stroke liners of large size are liable to crack in course of time at the breech end if the flange is unduly heavy The subject of the

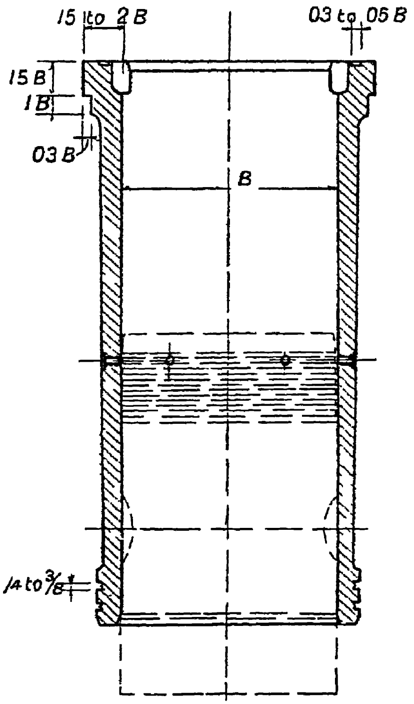


FIG 98

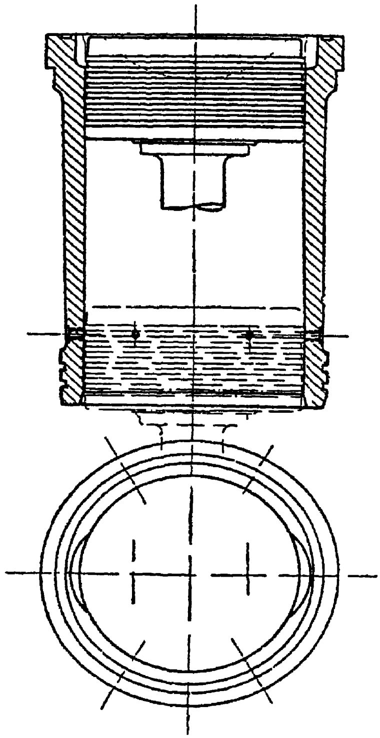


FIG 99

conduction of heat through the walls of the combustion space and the stresses induced thereby is of great importance in connection with large engines particularly and will be considered later

Typical liners for four stroke engines of the trunk and cross head types respectively are shewn in Figs 98 and 99 With the latter the piston is only of sufficient length to carry the rings and the length of the liner is determined by the position of the bottom ring at the bottom of the stroke With the trunk engine the liner must be long enough to embrace a sufficient

length (about equal to the bore) of the parallel part of the piston when at the bottom of its stroke in order to avoid a piston knock at the bottom dead centre It is therefore necessary to determine the clearance volume and complete the design of the piston before the length of the liner can be fixed finally

The bore is usually parallel with four stroke engines and slightly barrelled in way of the ports in two stroke engines to allow for the restraints which are inevitably placed at that position against free expansion of the liner Probably the best bore is produced by finishing with a reamer in a vertical machine Grinding is frequently adopted but there is a question if this process does not to some extent destroy those properties of cast iron which facilitate good lubrication The outside surface is frequently left unmachined in competitive work and there is probably no serious objection to this practice for four cycle work For two cycle engines it seems reasonable to take advantage of the increased heat conductivity obtain able by removing the skin

Strength of Liners —The upper end of the liner is subject to a working pressure of about 500 lb per sq in and the thick ness at this part measured under the heavy top flange may be found by the following formula which represents average practice for substantial engines —

$$\text{Thickness} = 0\ 08\ \text{bore} + \tfrac{1}{8}$$

The working stress being about 3000 lb per sq in in the case of a 30 in cylinder and less in smaller sizes explosions at starting etc may nearly double this stress occasionally Unfortunately the available information on the effects of repeated stress is not sufficiently complete at present to enable one to say definitely whether or not these excesses of stress have any influence on ultimate failure by fatigue but the writer is inclined to believe (on the strength of such evidence as has come before his notice) that the elimination of these occasional excess pressures would not enable any substantial reduction of thickness to be effected with the same margin of safety

On account of the diminution of pressure on expansion the liner may be tapered to a thickness of about 0 04 bore at the open end

The breech end of the liner requires to be reinforced by a

heavy flange to avoid distortion due to the pressure of the cover on the spigot joint Proportions are given in Fig 98

Points of Detail —The difficulty of accommodating the valves in the limited space available in the cover of a four stroke engine usually renders it necessary to make recesses in the top of the liner to clear the air and exhaust valve heads (see Fig 98) Four or more tapped holes are provided in a circumferential line round the liner to accommodate the lubricating fittings these being drilled when the liner is in position in the jacket The holes are located at about the level of the second piston ring (counting from the top) when the piston is at the bottom dead centre The fittings themselves will be described later The water joint between liner and jacket at the lower end may be made by one or more rubber rings The joint between cover and liner may be of copper or asbestos compositions

Two stroke liners are complicated by exhaust and some times air ports (see Fig 102) In the earlier designs the bars between the latter were always provided with water passages which introduced difficulties in manufacture and the value of which seems doubtful and these are now frequently omitted The fitting surfaces at this point are preferably ground to minimise chance of leakage as the high temperature prohibits the use of rubber packing rings

Cylinder Jackets —A simple and effective form of jacket for a four cycle engine is shewn in Fig 95 and in this example the jacket takes the pressure pull without the assistance of stay bolts The chief points to be observed are —

(1) A heavy flange at the top to carry the liner and to enable the tensile forces concentrated at the studs to dis tribute themselves uniformly round the jacket without producing high local stresses

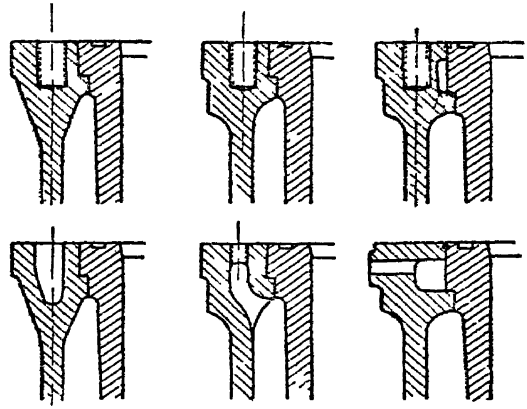
(2) A nearly plain cylindrical barrel as nearly as possible in line with the pitch circle of the cover studs and provided with sludge doors bosses for lubricating fittings and a bracket for supporting the cam shaft bearings

(3) A circular flange at the bottom for securing to the crank case

The remarks *re* tensile forces under heading (1) apply here also but to a less degree as the studs are pitched closer to gether than would be feasible on the cover On these consider

ations the thickness of the jacket for equal strength should taper gently towards the middle and the form shewn in the figure is the practical compromise Some of these points will be considered in greater detail

Top Flange of Cylinder Jacket —In small engines this may be solid but with larger sizes say from 15 in bore and upwards difficulty is sometimes experienced in obtaining sound metal at this point and coring of the flange between the studs is resorted to in order to accelerate cooling in the mould Different constructions are shewn in Fig 100 Schemes A and



C B A

FIG 100

B have the additional advantage of increasing the cooling surface Where four cycle engines are concerned the importance of this consideration is probably negligible Scheme B requires a water outlet connection between each stud if air pockets are to be avoided On the other hand the expense of coring is less than with scheme C

Barrel of Cylinder Jacket —This is sometimes conical instead of cylindrical and in this case it is reasonable to provide a vertical internal rib under each stud to discount the additional stress involved Consideration of manufacturing costs and of the good appearance of the engine rule out of court any form of external ribbing The brackets supporting the valve gear take many forms in different designs That shewn in Fig 95 is the modern form and considered in conjunction with the gear it supports appears to combine most advantages including that of elegance

Bottom Flange of Jacket —If four staybolts are provided for each cylinder these may conveniently be used to secure the latter to the crank case The concentration of the tensile load at four points necessitates a heavy flange preferably of box form as shewn in Fig 101 The corners of this flange being each subject to a load of one quarter of the maximum working pressure load deserve attention in the form of a calculation of the bending stress involved A plain square shape would appear to be preferable to some of the more elaborate shapes which have occasionally been used the flat sides lending themselves well to the provision of facings for various purposes

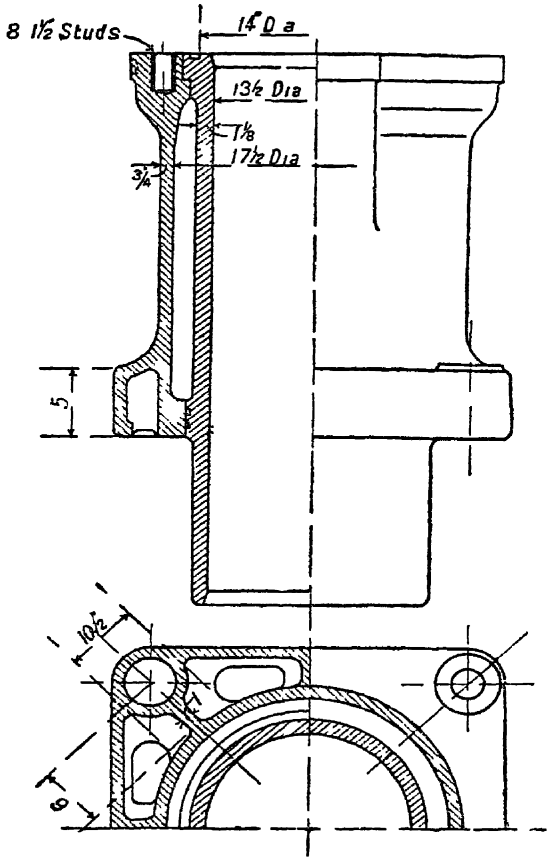


Fig 101

Frequently the flange is spigoted into the top of the crank case but as this involves an unnecessary machining operation on the latter and makes cylinder alignment more difficult the better practice is to core the aperture in the crank case sufficiently large to allow for adjustment of the position of the cylinder and to locate the latter by means of two steady pins

Strength of Four Stroke Cylinder Jackets —The considerations of strength which enter into the design of a cylinder jacket are illustrated by the following check calculations relating to the cylinder shewn in Fig 101

Bursting stress in liner

$$=\frac{500 \text{ (lb per sq in)} \times 6\ 75}{1\ 125}=3000 \text{ lb per sq in}$$

Nominal pull in each cover stud $= \frac{500 \times 0\ 784 \times 14^2}{8} = 9650$ lb

Permissible nominal load for $1\frac{1}{2}$ in stud according to table on page 129 9300 lb

Maximum working pull in jacket $= 0\ 784 \times 13\ 5^2 \times 500 =$ 71 500 lb

Tensile stress in jacket $= \frac{71\ 500}{\pi \times 18\ 25 \times 0\ 75} = 1670$ lb per sq in

Owing to the peculiar shape of the bottom flange the calcu lation of its strength presents a difficulty which is easily evaded by substituting for the actual section a simpler one of obviously inferior strength

Nominal load at each corner 71 500 lb $-4 = \sim 18\ 000$ lb

Moment from centre of bolt to jacket wall $= 18\ 000 \times 9$ in lb

Modulus of hypothetical section —

$$z = \left(\frac{10\ 5 \times 5^3}{12} - \frac{9\ 5 \times 3\ 5^3}{12}\right) - 2\ 5 = 30\ 2 \text{ in }^3$$

$$\text{Stress} < \frac{18\ 000 \times 9}{30\ 2} \quad \text{i e} \quad < 5\ 400 \text{ lb per sq in}$$

In view of the unfavourable assumptions this is probably not excessive for first class cast iron

Jackets for Two Stroke Engines —The necessity for provid ing exhaust passages or belts and in some cases passages for scavenge air as well introduces considerable complication into the design renders the stresses in certain parts more or less in determinate and makes greater demand on the skill of the manufacturing departments in comparison with that required by four cycle construction

Referring to Fig 102 it will be seen that the exhaust belt interrupts the vertical line of the jacket wall and if the latter has to carry the main tensile stresses internal ribbing becomes a necessity The arrangement shewn is perhaps as good as any but the attachment of ribs to the exhaust belt has a restraining influence on the temperature expansion of the latter which can only result in mutual stresses It appears however that these are not very serious as cylinders which have failed in other respects have remained intact at this point Fig 103 shews a construction in which a good attempt is made to secure continuity of the vertical wall of the jacket Either of these systems i probably satisfactory for cylinders of medium size Large cylinders however (and this applies to

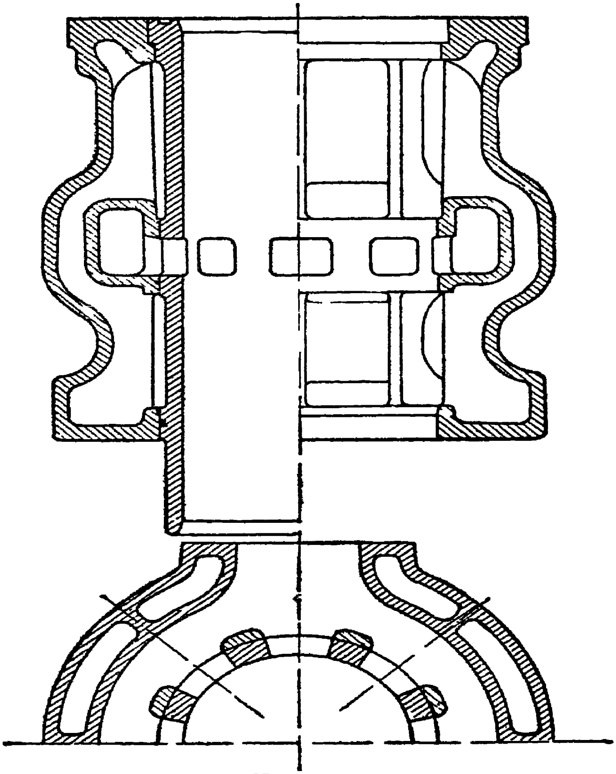


FIG 102

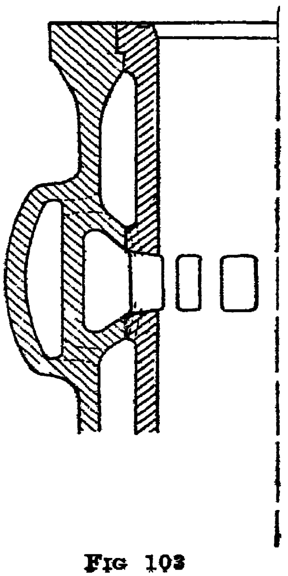


FIG 103

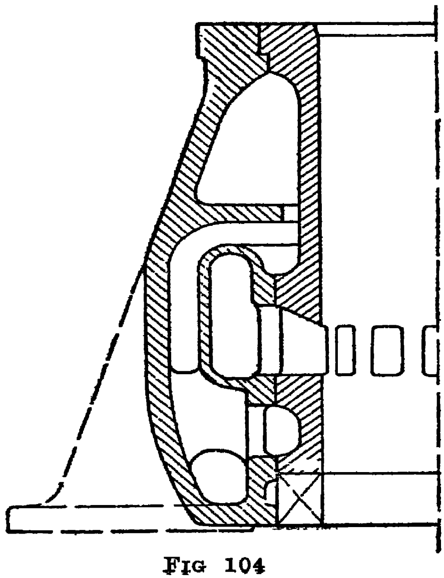


FIG 104

other parts as well) are known to be subject to greater temper ature differences than smaller ones (though not to the extent sometimes suggested) and the leading designers have had recourse to other expedients when faced with the problem of constructing cylinders of large size

In Fig 104 the jacket wall may be described as similar to a honey pot in shape and of abnormal thickness to allow for the bending stresses caused by the curvature of the walls and the fact that the tensile supporting forces are localised at two feet

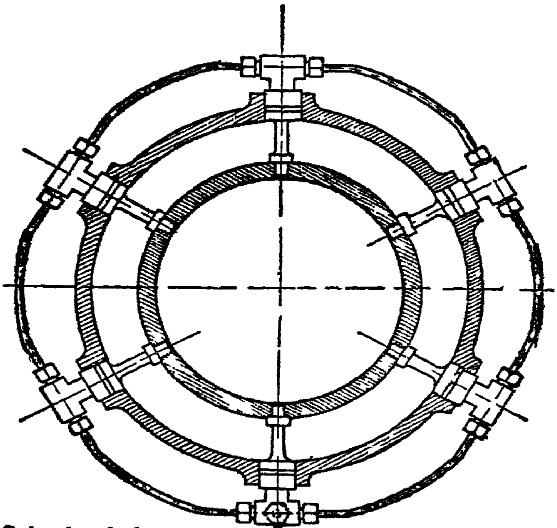


FIG 105 (See p ge 151)

The exhaust belt is of relatively thin metal with compaiatively small support from the walls It will be evident that the strength of the jacket is very slightly influenced by the exhaust belt and that the latter is free of all but temperature stresses This construction therefore attains a good approximation to the correct allocation of the respective duties of jacket and exhaust belt

As disadvantages may be cited abnormal weight of cylinder and the difficulty of casting a cylinder involving widely different thicknesses of metal

Another and perhaps better way out of the difficulty is to connect the cylinder cover to the bedplate by means of stay

bolts thus relieving the jacket of all stresses except those induced by temperature differences The jacket in this case virtually hangs from the cylinder cover and only requires to be attached thereto by studs proportioned to a load based on the cylinder pressure and the annular area lying be tween the cylinder bore and the spigot at which the cover joint is made The upper flange is preferably made fairly substantial but other thicknesses may be made a practical minimum

Cylinder Lubricating Pump

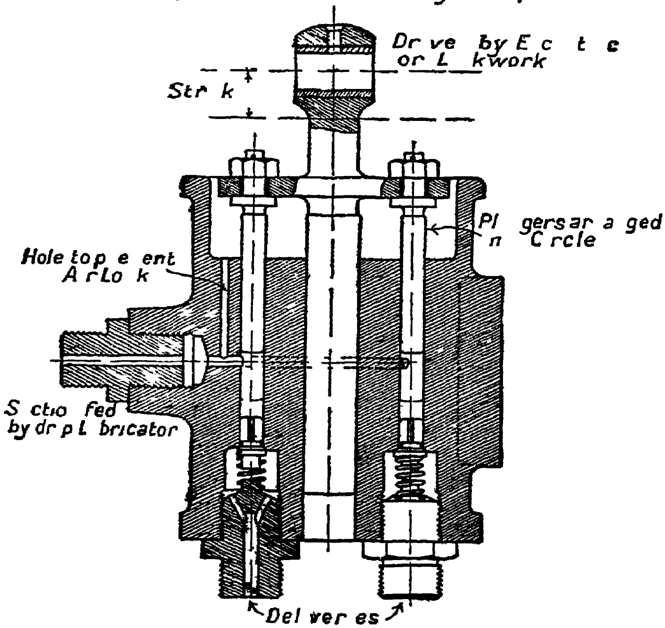


FIG 105 (See page 151)

Cylinder Lubrication —The problem of cylinder lubrication in Diesel Engines consists in effecting uniform distribution of minute quantities of oil The quantity of oil admitted must be the minimum necessary to effect satisfactory lubrication as the oil cracks in service leaving a gummy deposit which in course of time causes the piston rings to stick Under favour able conditions this may be several months even a year Every drop of superfluous oil reduces this period hence the importance of uniform distribution so that every part may have sufficient but none a superfluity These conditions are

best secured by a separate controllable feed to each of about six or eight points round the circumference of the cylinder A typical lubricating fitting is shewn in Fig 105 (pages 149–151) and the point to be observed is that the fitting must adapt itself to slight relative movement between the liner and jacket The small hole at the end which leads to the surface of the liner reduces to a minimum the chances of the fitting becoming choked with carbon With forced lubricated engines in which the cylinder is not isolated from the crankpit it frequently happens that more than sufficient oil reaches the cylinder apart from any arrangements made for the purpose In this case the problem may be to devise scraper rings vent holes or other devices to remove the superfluous oil

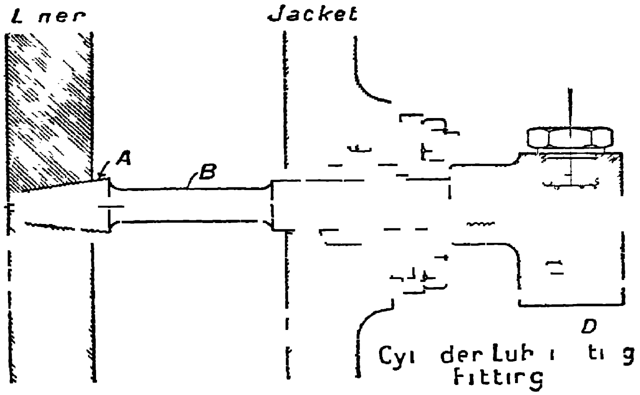


FIG 105 (See also pages 149 150)

Cylinder Covers —Owing to a considerable number of failures in service and difficulties experienced in manufacture cylinder covers for both four and two cycle Diesel Engines have come to be regarded as difficult pieces of design and it may perhaps be instructive to review the subject in a more or less historical manner

The earlier type of four cycle cover is shewn in Fig 106 from which it will be seen that the internal coring is complicated and that a few core holes of small diameter only are provided to vent the core in the mould In spite of these disadvantages such covers have given good service when made by the most skilful of continental manufacturers Dismissing for the moment the question of manufacturing costs these covers have the following shortcomings —

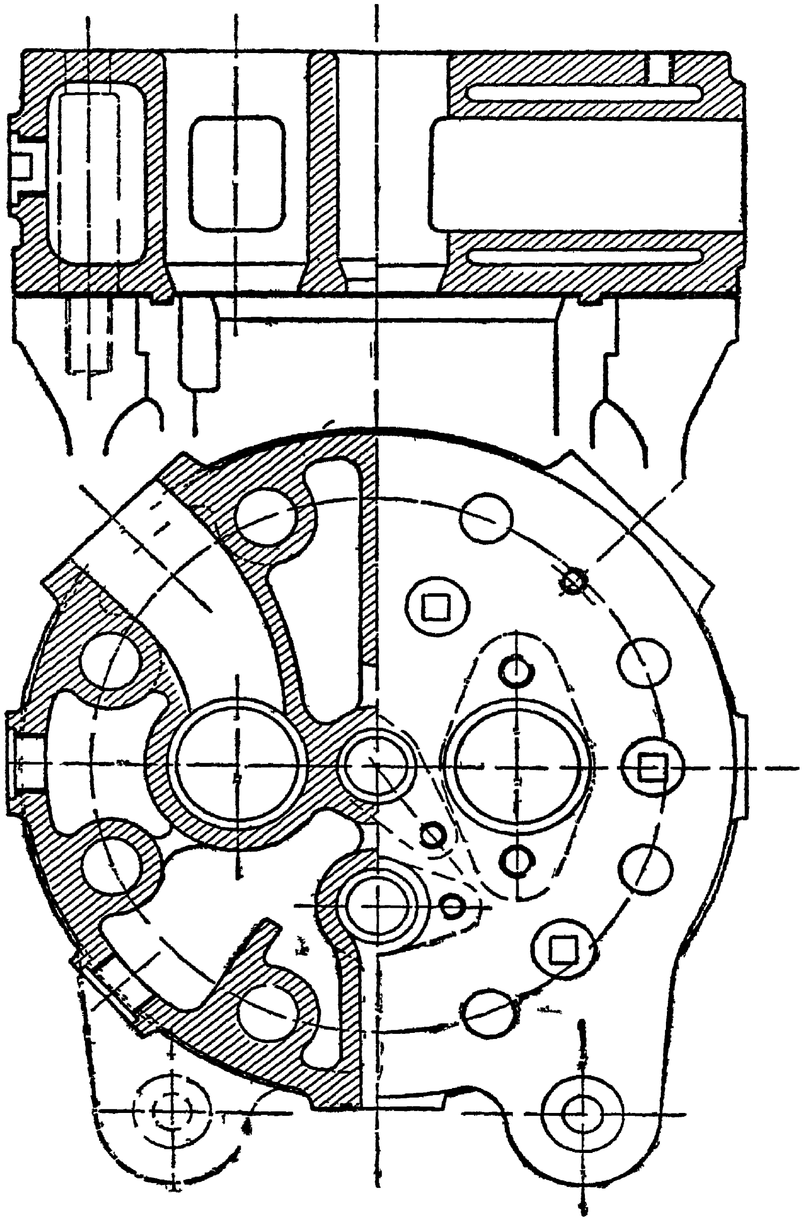


Fig 106

(1) The thin walls of uncooled metal between the recesses for fuel and exhaust valves are liable to crack on over loaded engines

(2) The hot exhaust passage is too rigid to permit of much expansion and leads to cracking of the bottom plate

(3) The small core holes give poor access to the interior for purposes of cleaning away accumulated scale

Assuming first class foundry work the two latter con idera tions are perhaps the most important Modern development is on the following lines —

(1) The provision of large doors which serve the double purpose of providing good access for cleaning and affording better support and venting for the core when casting

(2) Elimination of all internal ribs as experience seems to shew that the tubular walls provided to accommodate the valve casings provide all requisite support between the top and bottom plates

(3) Using brass or steel tubes expanded into the recesses for the fuel valve and holding down studs

(4) The use of square instead of conical seats for the valve casings

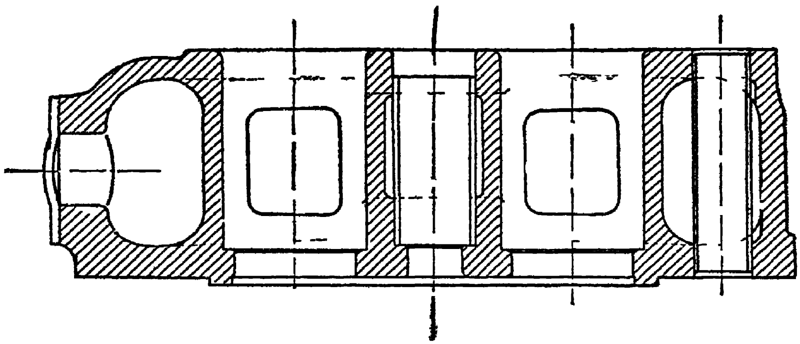
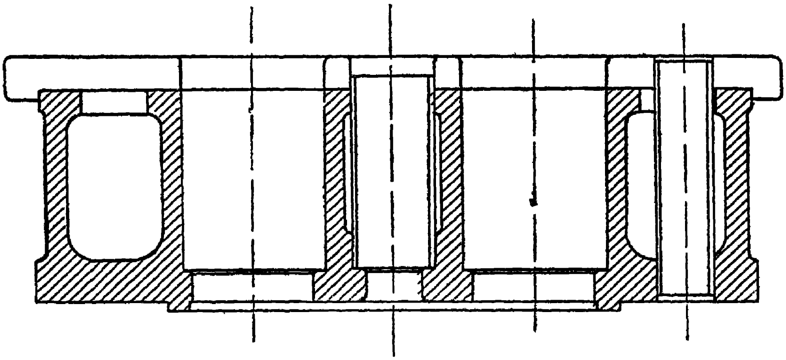


FIG 107

A cover designed on these lines is shewn in Fig 107 Some makers have simplified the question of casting at the expense of introducing extra machining and fitting operations by making the top plate a separate piece (see Fig 108)

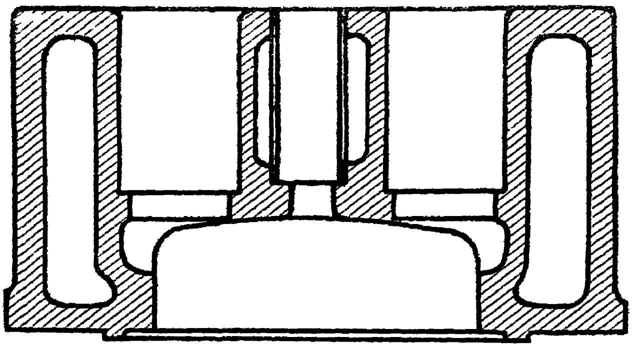
Another innovation which is becoming increasingly common is to place the fuel valve off centre This arrangement enables

the coolıng space around the fuel valve to be ıncreased but too great a dısplacement of the fuel valve from the cei tre posıtıon necessıtates a specıal shape of combustıon space



Fɪɢ 108

Some modern developments ın fouı stroke cylınder cover constructıon aıe shown dıagrammatıcally ın Fıgs 109 to 112 That of Fıg 109 follows closely on tradıtıonal lınes wıth the exceptıon that the lower plate ıs dıshed upwards to allow more freedom foı expansıon and to gıve more ıoom foı the



Fɪɢ 109

valves Fıg 110 represents a combıned cover and lıner as used by the Werkspoor Co Fıg 111 shows the kınd of con struction adopted ın some of the recent Burmeıster and Waın engınes In thıs case the lıner ıs secuıed to the cover by a flange and setscrews Fıg 112 shows a cover fitted wıth a water cooled pad to receıve the heat whıch would otherwıse reach the cover proper

Points of Detail —Owing to the large recesses for the valve cages a four cycle cover is relatively weak considering the amount of metal in it and on this account all stud holes should be well bossed under and all inspection openings well reinforced by compensating rings like a boiler

The under face of the cover is machined all over but on the

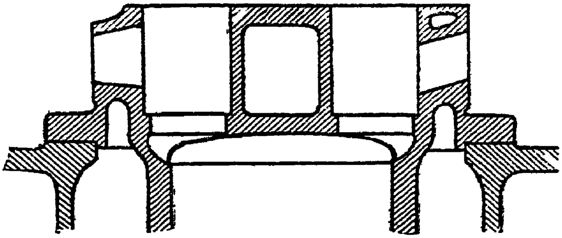


FIG 110

top face machining is sometimes restricted to those parts which are occupied by valves etc This enables the corners to be given a liberal radius which in addition to improving the appearance facilitates moulding (see Fig 107) From all considerations all internal angles should be well radiused

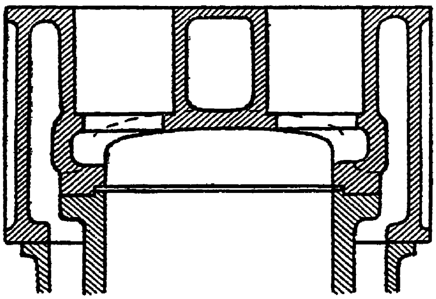


FIG 111

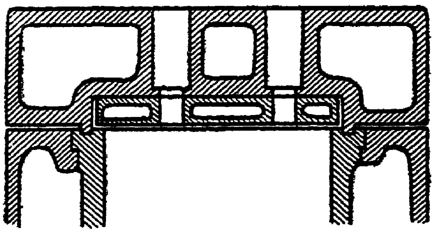


FIG 112

Water is led to the cover by one of two methods —

(1) By one or more tubular fittings screwed into the top of cylinder jacket and passing through holes in the under face of the cover (see Fig 113) With the type of jacket shewn in Fig 100B it is desirable to fit one such fitting between each pair of cover studs

(2) By means of an opening in the side of the cover (see Fig 114)

Whatever means be adopted it is advantageous to fit internal pipes or baffles to encourage flow towards the fuel valve as accumulation of deposit at this point is to be avoided at all cost It is usual to arrange the outlet above the exhaust branch as stagnation at this point is also undesirable

Proportions of Cylinder Covers —The depth of a four stroke cover generally works out to about 0 7 of the cylinder bore the limiting factors being the size of the exhaust passage and the water space around it The former should be at least equal in area to the exhaust valve at full lift The passage starts by being rectangular in shape at the valve end and gradually becomes circular at the outlet where the diameter is

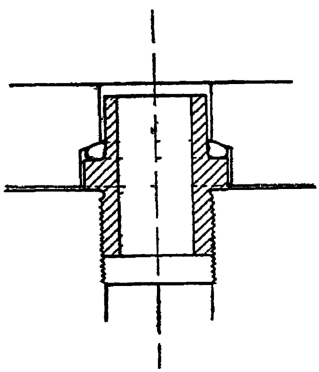


FIG 113

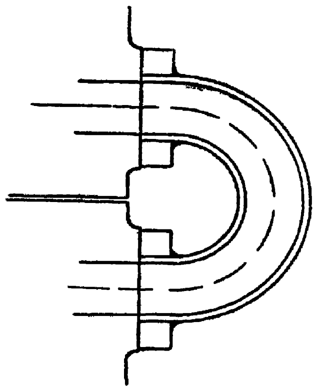


FIG 114

about 0 31 of the cylinder bore The same applies to the air inlet passage The thicknesses of metal vary considerably in different designs and the proportions shewn on the sketches represent average practice The bottom plate should either be fairly thick as shewn (small engines) or well supported internally round the spigot in larger sizes to prevent caving in

Strength of Four Stroke Cylinder Covers —The system of loads acting on the cover comprises the tightening stresses of the studs the reaction at the spigot joint and the gas pressure on the lower plate The effect of such a system is to produce tensile stress in the top plate and compression on the bottom Considering the relative weakness of cast iron in tension and the fact that cracks in the top plate are of very rare occurrence it would appear that covers proportioned in accordance with average practice have a good margin of safety so far as pressure

stresses are concerned In view of a few isolated failures or rather as a matter of principle the strength should be subject to calculation Owing to the uncertainty as to actual conditions the method of calculation detailed below must be considered comparative rather than absolute

The assumptions underlying the method are as follows —

(1) That the severest conditions of stress are due to a cylinder pressure of 1000 lb per sq in due to pre ignition careless starting or otherwise and that this pressure causes the cover to lift to such an extent that the reaction at the joint is eliminated

(2) That the stress is uniform across a diametrical section in the case of a cover of constant depth and proportional to the distance from the neutral axis of the section in the case of a cover of varying depth This is not correct but probably involves approximately equal percentage of error in different cases

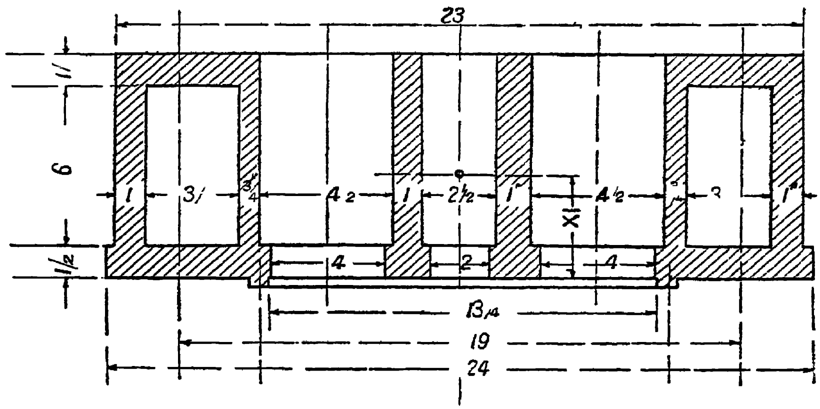


FIG 115

Example Referring to Fig 115 shewing the weakest section passing through the recesses for the air and exhaust valves the section modulus is 150 in ^{3}

Considering the forces to the right or left of this section we have —

(1) A downward force at the stud circle equivalent to a pressure of 1000 lb per sq in over half the circular area extending to the joint spigot viz $784 \times 13\ 25^2 \times$

1000 − 2 = 69 000 lb This may be considered to act at the centre of gravity of the pitch semicircle that is at a distance of $9\ 5 \times 2 - \pi = 6\ 02$ in from the section under consideration

(2) An equal and opposite force on the under side of the cover acting at the centre of gravity of the semicircular area extending to the joint spigot i e at a distance of $\frac{6\ 625 \times 4}{3\pi} = 2\ 8$ in from the centre

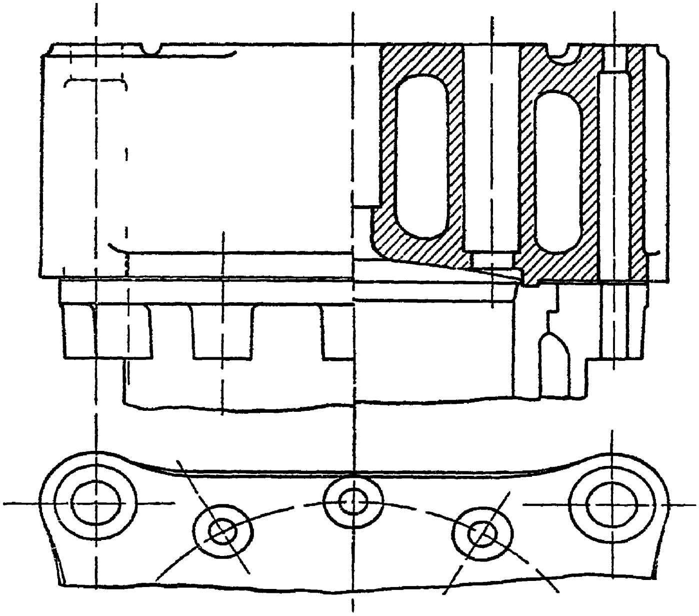


Fig 116

The stress is therefore —

$$\frac{69\ 000\ (6\ 02 - 2\ 8)}{150} = 1500 \text{ lb per sq in}$$

Putting the above into the form of a rule —

$$f = \frac{1000\ R_1^2\ (R_2 - \frac{2}{3}\ R_1)}{z}$$

Where 1000 = Assumed maximum pressure
f = Stress in lb per sq in
R_2 = Radius of stud pitch circle in in
R_1 = Inside radius of joint ring in in
z = Section modulus in in ^{3}

Two Stroke Cylinder Covers —Where port scavenge is adopted the cover has only to accommodate the following fittings —

(1) Fuel valve (3) Relief valve (if fitted)
(2) Starting valve (4) Indicator tube fitting

As all the above are relatively small the casting of the cover is much simpler than that for a four stroke engine Fig 116 shews a cover of this type arranged for four staybolts This type of cover is not suitable for large highly rated

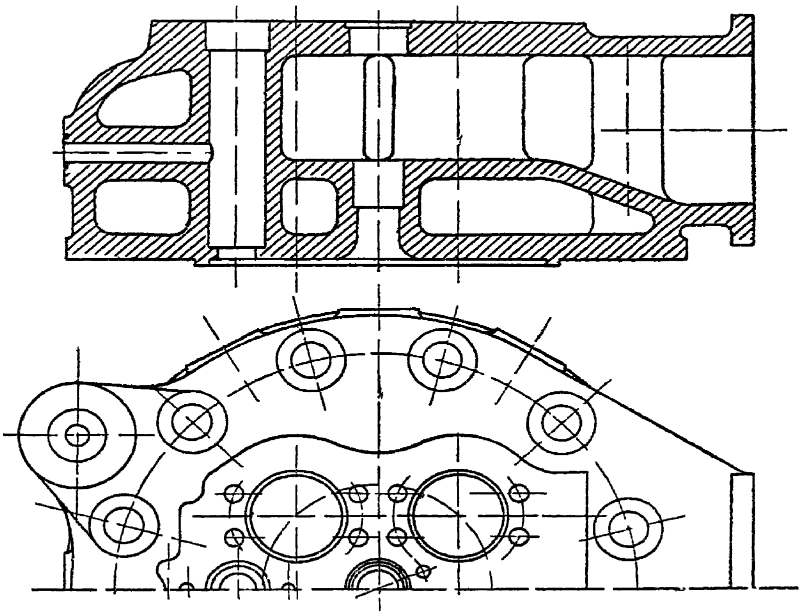


Fig 117

two stroke engines on account of the liability of the bottom plate to crack under the influence of heat stresses

When valves in the cover are employed for scavenging purposes the construction depends on the number of valves If two scavenge valves are used the cover may be similar to that of a two stroke engine This arrangement seems to have fallen into disuse no doubt on account of the difficulty of securing adequate valve area and efficient scavenging

Fig 117 shews a cover designed to accommodate four scavenge valves It will be noticed that the interior is divided

by a horizontal diaphragm separating the air space and the water jacket It appears that this diaphragm and the tubular connections to the bottom plate impose too great restrictions on the expansion of the latter and fractures have been frequent (with both cast iron and cast steel) so that this type

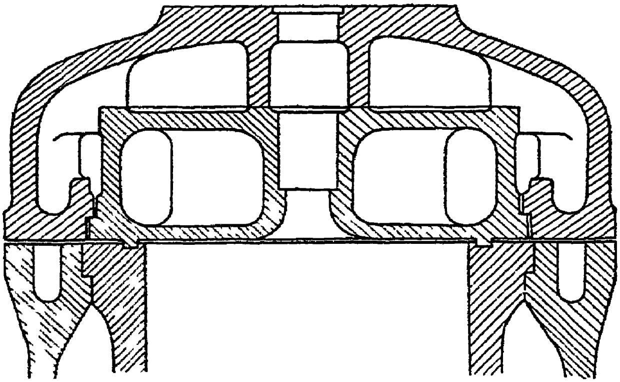


FIG 118

of cover as hitherto designed must be considered a failure The writer understands that a modification of this design (patented by Mr P H Smith) shewn in Fig 118 has proved satisfactory and failures have been greatly reduced in frequency Apparently the additional depth of the water jacket

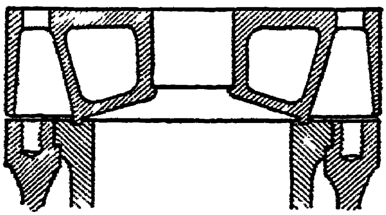


FIG 119

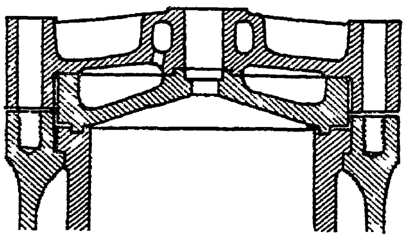


FIG 120

and correspondingly increased freedom of expansion minimise temperature stresses and the support afforded by the external shell keeps the bending stresses to a moderate figure

Some recent types of two stroke cylinder covers are shewn diagrammatically in Figs 119 to 121

The first represents the Sulzer construction in which the fuel, starting and relief valves are accommodated in a central

water jacketed cage The bottom plate is comparatively thin and ceases at the spigot The symmetry of the casting and the flexibility of the walls are favourable conditions from the point of view of immunity from heat stresses Fig 120 shows a scheme used on the White two stroke engine This appears to aim at freedom of expansion and easy replacement of the bottom plate should fracture occur Fig 121 shews another application of the cooled pad idea These examples illustrate the diversity of the designs which have arisen very largely with a view to reducing the liability to failure on account of temperature stresses which are considered in the next chapter

The Shape of the Combustion Space —It has often been

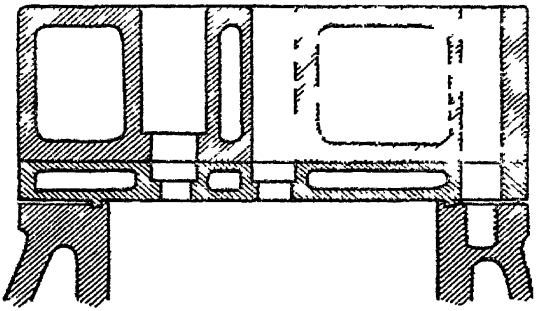


Fig 1 1

said that the ideal shape for the combustion space of an internal combustion engine is that of a sphere since for a given volume this shape offers the smallest surface for the dissipation of heat Such an ideal cannot be realised in practice as a combustion space which is spherical when the piston is on top dead centre is no longer so when the piston has moved outwards on the expansion stroke

This consideration applies particularly to the Diesel Engine since the maximum temperature of the Diesel cycle is only attained after a certain fraction of the expansion stroke has been performed For this reason the spherical combustion space shewn in Fig 122 1 would probably shew little if any advantage over the flat shape of Fig 2 so far as heat loss is concerned though the distribution of fuel and resulting effi ciency of combustion would probably be better with Fig 1

So far as the writer is aware no such shape as that indicated

in Fig 1 has been used but the approximation shewn in Fig 3 has been adopted in two or three makes of small and moderate sized engines One advantage claimed for this shape is the turbulence caused by the expulsion of air from the annular space between the piston and the cylinder head on the compression stroke The reverse flow on the expansion stroke is probably responsible for an increased rate of heat flow to the walls of the cover

The first essential in a Diesel Engine is good combustion and this can only be secured if the fuel is well distributed in

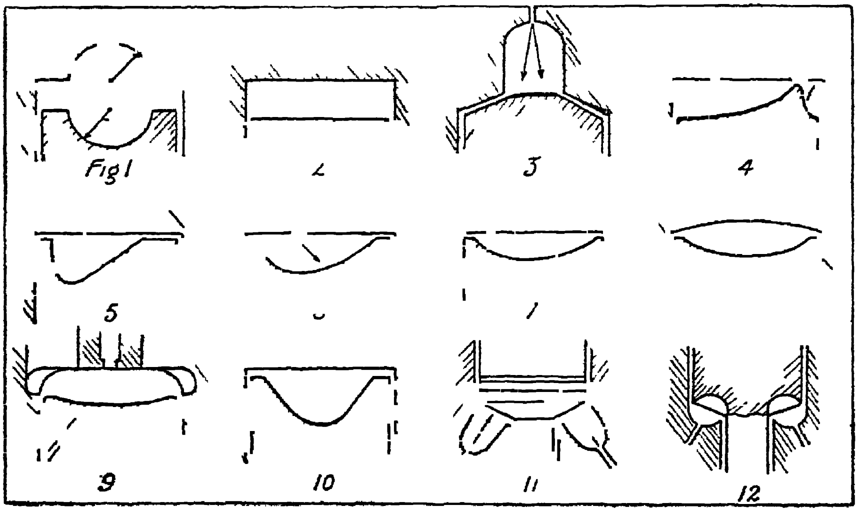


Fig 122

the first instance The combustion space should therefore be fairly compact at the dead centre The shape shewn in Fig 4 for example would render impossible good combustion at full load on account of the dead air at A to which the fuel could not find access On the other hand the peculiar shape shown in Fig 5 is capable of excellent results provided the fuel blast is properly directed as shewn A similar shape (Fig 6) sometimes gives inferior results if the fuel valve is offset too far from the centre Moderate offsetting of the fuel valve seems to have little if any effect with large cylinders

The earliest four stroke engines were provided with flat clearance spaces as in Fig 2 Better results are obtained with the plano convex shape (Fig 7) or the convexo convex

shape of Fig 8 The flat shape is however well suited to opposed piston engines and perhaps to single piston engines having a stroke to bore ratio of about 2 1 and upwards since the combustion space is then fairly deep

Sometimes the maximum diameter of the combustion space is made in excess of the cylinder bore in order to obtain more room for the valves as in Fig 9 In these cases care must be exercised to avoid pockets of dead air Some designers aim at providing a considerable depth of air in the direction of the fuel blast as in Figs 3 and 10 The maximum temperature of the piston is probably reduced by this construction Some form of the pocket type of combustion space would appear to be the only feasible one for double acting engines (Figs 11 and 12)

The requirements of the combustion space in relation to good combustion may now be summarised as follows —

(1) Compact shape of space at top dead centre
(2) Direction of fuel blast approximately through the C G of the combustion space if two fuel valves are fitted each fuel blast should be directed at the C G of a symmetrically placed half space
(3) Either (*a*) normal impingement of fuel blast
or (*b*) injection into a considerable depth of air

Literature — Diesel Engine Cylinder Dimensions —Article in *Engineering* September 5th 1913

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CHAPTER IX

TEMPERATURE STRESSES

Introductory —It is fairly generally recognised that the stresses in the cylinders of internal combustion engines due to temperature differences are at least as important as those due to pressure at any rate in the larger sizes of engine and that the magnitude of these stresses sets an upper limit to the size of cylinder which is possible with any given method of construction

Such considerations apply with even greater force to cylinder covers and pistons but it is not necessary to agree with those who maintain that the maximum size of cylinder possible or the maximum power which can safely be developed in a cylinder of given bore and stroke is necessarily limited by the materials available

It is true that every substantial advance in size and power rating accentuates the heat stress problem but there is no reason to suppose that such problems cannot be solved satisfactorily Some of the newer constructions which have been developed to overcome temperature difficulties have already been noticed and it can hardly be doubted that demand will be met by supply

In the present chapter the general heat flow problem as it affects Diesel Engine Design will be considered in detail with a view to throwing some light on the principles involved

Temperature difficulties in Diesel Engines may be classified in three categories —

(1) Local overheating as for example of exhaust valve heads piston crowns rings and liner surfaces

(2) Local temperature differences as for example the temperature difference between the gas and water sides of a liner or cylinder cover plate

(3) Extensive temperature differences as for example the temperature difference between the centre and the edge of an uncooled piston crown

The troubles coming under the first category are avoided or minimised by either avoiding uncooled surfaces altogether or by making the uncooled part of a special material capable of performing its function at a high temperature

The stresses which arise on account of the second category can only be kept within safe limits by either limiting the heat flux to which the surface is subject or else limiting the thickness of metal through which the flux penetrates

Extensive temperature differences (3) are in general only productive of temperature stresses in so far as the temperature differences refer to different parts of the same casting In these cases the stresses are to be kept within safe limits by adopting forms which admit of free expansion A study of the problems which arise out of these considerations must clearly start with an estimate of the distribution of heat flux to the various members which constitute the walls of the cylinder volume

Heat Flow to the Jackets—The heat received by the cylinder walls may be regarded as transferred thereto by convection and radiation Any element of pure conduction may presumably be neglected except in so far as the process of convection culminates in the conduction of the convected heat through a thin layer of stagnant gas in contact with the walls The relative importance of the two processes of radiation and convection in transferring the heat lost to the jacket is not known with any certainty but a number of important facts have emerged as a result of the investigations some of which are referred to at the end of this chapter among these may be mentioned the following —

(1) The heat lost by convection is increased by increasing the state of turbulence before and during ignition (Hopkinson and Clerk) This fact helps to account for the observed fact that increasing the piston speed beyond very moderate values does not very materially reduce the percentage of heat rejected to the jacket Apparently the increased air speed through the inlet valve results in increased turbulence which tends to speed up the loss of heat

(2) A mass of flaming gas is semi opaque semi transparent to its own radiations That is to say of the total radiation emanating from a small element of flame a certain part (usually a small fraction in cylinder volumes of ordinary size)

is absorbed in passing through the outer layers (Callendar) The practical bearing of this fact is open to different interpretation in the present state of knowledge According to one school of thought the semi transparency of flame is fatal to large cylinders on account of the augmented radiation flux which results from increasing the depth of the flame On the other hand it may be argued that the partial opacity of flaming gas must lead to the rejection by radiation of smaller percentages of the heat liberated by combustion in cases of large cylinders as compared with small ones Both arguments are unsound since both ignore the duration of the combustion period and the question whether the total radiation is simply a function of temperature and time or whether it depends on the amount of heat evolved by combustion Fortunately the heat flow question does not depend on the solution of these problems The total heat received in all ways by the various elements of the cylinder walls can be measured approximately by various more or less direct means These measurements indicate that the jacket loss expressed as a percentage of the total heat supplied is nearly independent of the absolute size

(3) The intensity of radiation is proportional to some power (3 to 4) of the absolute temperature This fact helps to account for the fact that indicated efficiencies tend to fall off after a certain critical maximum temperature is exceeded In what follows we shall only be concerned with the Diesel Cycle (four stroke or two stroke) employing a normal full load M I P of about 85 to 105 lb per inch

The water jacketed parts of a Diesel Engine comprise some or all of the following —

(1) Cylinder liner
(2) Cylinder cover
(3) Exhaust valves and cages
(4) Piston
(5) Air compressor
(6) Intercoolers
(7) Exhaust manifold
(8) Crosshead guides
(9) Oil coolers

Here we are only concerned with items (1) to (4) inclusive which on the average may be taken to account for about 25% of the total heat supplied reckoned on the lower calorific value of the fuel consumed at about full load This will be referred to below as the total jacket heat

Whilst water cooled valves and cages absorb quite an

appreciable percentage of the jacket heat it is doubtful if the cooling of these parts materially affects the total jacket heat since the exhaust gases are cooled thereby and are consequently in a condition to give less heat to the exhaust port in the cylinder head or cover The same argument applies to the piston Heat received by an uncooled piston is mainly con ducted to the cylinder liner which therefore receives more heat than it would otherwise if the piston were cooled Before considering the distribution of the jacket heat it may be remarked that the latter under approximately full load con ditions when expressed as a percentage of the total heat supplied shews remarkably little variation with such variables as piston speed mean indicated pressure stroke bore ratio absolute size etc within the limits of ordinary Diesel Engine practice

Cylinder Cover —The heat received by the cover of a four stroke engine consists of two parts —

(a) That which passes through the bottom plate
(b) That which passes through the exhaust valve port

For simplicity we suppose that the cover is of the traditional flat bottomed type Gibson and Walker have attempted to measure b in the case of a gas engine and found an exhaust valve and port loss of 8 to 11% of the total heat supplied or rather less than one third of the total jacket loss as here understood From the conditions of the experiment these figures must be over estimated Dugald Clerk arrived at a similar estimate based on Burstall s gas engine trials but this estimate is vitiated by the assumption that the heat passing through the cylinder cover face was equal to that received by the piston an assumption which Hopkinson has shewn to be very improbable in a gas engine Hopkinson s estimate of the exhaust port heat is about 6% of the total heat supplied in the case of a gas engine For Diesel Engines the writer is inclined to adopt the figure of 4½% of the total heat supplied or about 18% of the total jacket loss for reasons which appear below

In the first place items a and b together only account for about 35% of the jacket loss in the case of a four stroke Diesel Engine having a stroke bore ratio of about 1 5 Further more recent temperature gradient measurements in the cover plate of a four stroke Diesel Engine having a stroke bore ratio

of 1 indicate that the bottom plate in this case receives abou 28% of the total jacket heat If the stroke bore ratio wer increased to 1 5 the cover plate loss expressed as a percentage of the jacket loss would certainly be reduced and woulc probably be about 17% (see Fig 124) leaving 35 − 17 = 18% for the port loss As a first approximation we therefore adopt the following figures —

PERCENTAGE OF TOTAL JACKET HEAT

Stroke bore ratio	1 0	1 5
Cover plate	28	17
Exhaust port	18	18
Total for cover	46	35

It will be noticed that it is here assumed that the exhaust port loss is independent of the stroke bore ratio

In the case of two stroke engines there is no exhaust port in the cover and we are only concerned with the bottom plate

Pistons —Even with uncooled pistons the maximum crown temperature is low compared with that of the gases during the combustion and exhaust periods and it has been shewn that the heat taken up from the piston by the charge is small The net heat received by the piston crown is not therefore greatly influenced by the cooling means adopted Hopkinson has shewn that the heat dissipated from the underside of the crown of an uncooled gas engine piston has very little effect on the temperature of the latter He has also shewn how to calculate the amount of heat received by the piston from two temperature measurements at two different distances from the centre of the crown On these assumptions the piston of his $11^{1} \times 21$ gas engine received about 12% of the total jacket heat In Burstall s trials of a 16 × 24 gas engine the water cooled piston received about $16^{1}\%$ of the total jacket heat

With the four stroke Diesel Engine with square cylinder already referred to the measured piston heat amounted to 12% of the total jacket heat but in this case it is probable that owing to the relatively mild character of the oil cooling a certain amount of heat received by the piston was conducted to the liner The water cooled piston of a 15 4 × 15 75 two

stroke Diesel Engine absorbed about 22% of the total jacket heat

The water cooled piston of a 16 1 ×19 9 two stroke Diesel Engine received on the average about 36% of the total jacket heat This abnormally high figure is probably due to —

(1) The use of a convex piston crown
(2) The liberal extent of the piston jacket which extended half way down the skirt
(3) About a quarter of the total quantity of cooling water was passed through the pistons

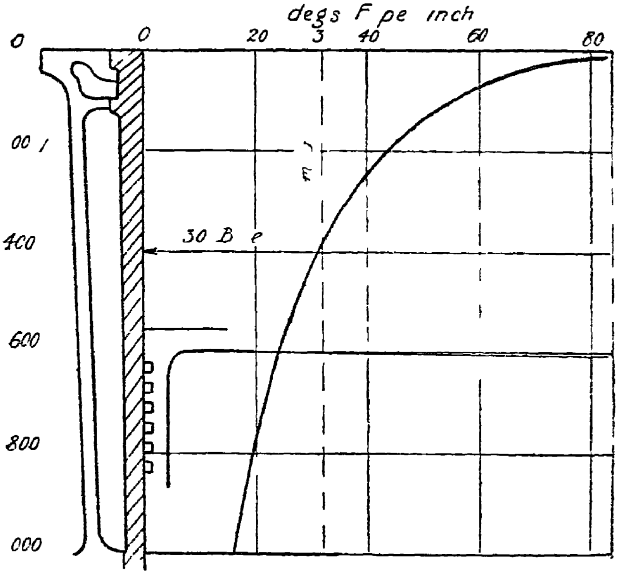

FIG 123

In this case the piston was probably extracting heat from the liner

In a modern two stroke engine with concave piston crown and stroke bore ratio of 1 6 the cylinder jacket received about 10% and the piston and cylinder cover about 6% each of the total heat supplied

The percentage of jacket heat absorbed by the pistons of two stroke engines may reasonably be expected to exceed that of four stroke engines on account of the exhaust taking place at the bottom of the cylinder with great velocity

Comparing these figures and confining attention to the heat actually passing through the piston crown it would appear that in four stroke engines the piston heat is rather less than the cover plate heat and that in two stroke engines those quantities are about equal assuming concave piston crowns in both cases

Cylinder Liners —In four stroke engines with stroke bore ratios of about 1 5 with uncooled pistons about 65 % of the

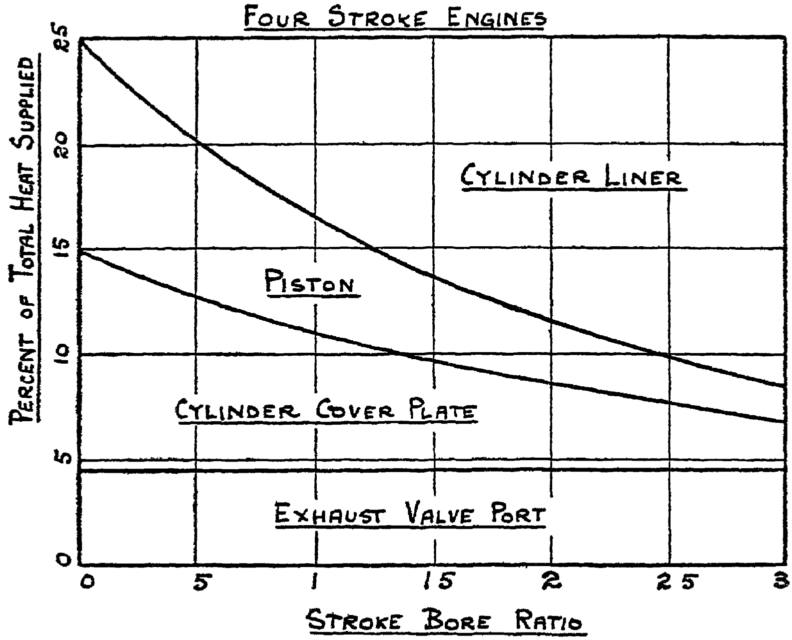

FIG 124

jacket heat is found in the cylinder jacket Deducting 16% for the piston this leaves 49% for the liner heat proper The thermal data given by Chaloner (*Motor ship* July 1920) in connection with a four stroke cylinder 20 85 ×20 85 enable the jacket loss to be estimated The temperature gradients are shown in Fig 123 from which it appears that the mean temperature gradient is about 32° F per inch and the mean flux is therefore —

$$32 \times 12 \times 26 = 10\,000 \text{ B T U per ft}^2/\text{hr}$$

Since the conductivity of cast iron is about 26 in ft deg F

houı unıts The total area of the lıneı ıs about 19 2 ft ² so the total heat flow ıs —

10 000 × 19 2 = 192 000 B T U per hour

The power of the cylınder ıs 285 B H P and reckonıng on a fuel consumptıon of 0 42 lb per B H P hr the total heat supplıed works out at —

0 42 × 18 000 × 285 = 2 150 000 B T U /hı

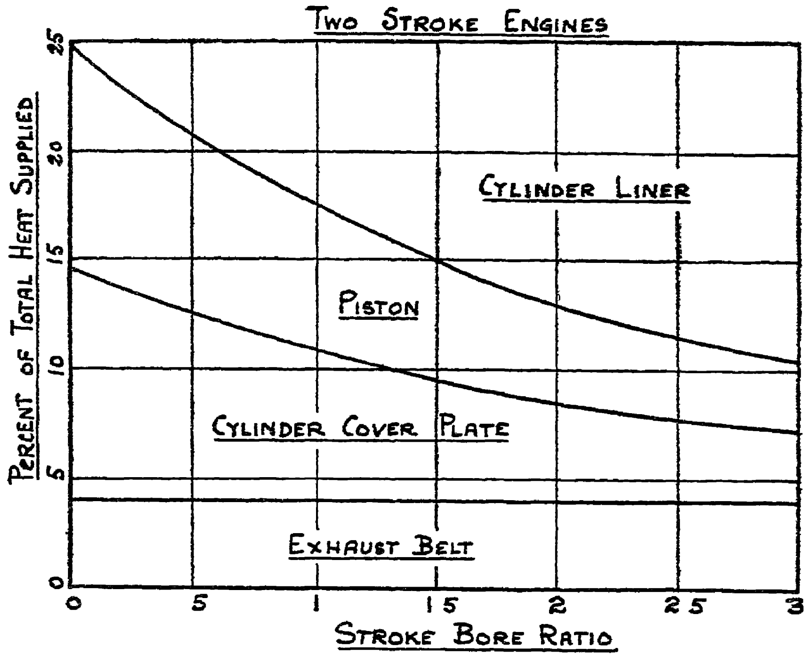

Fig 125

The lıneı heat ıs therefore 8 9% of the total heat supplıed or about 35% of the total jacket heat

Heat Flow Dıagrams —On the basıs of the foregoıng figures and the consıderatıon of lımıtıng cases Fıgs 124 and 125 have been drawn up for four and two stroke engınes respec tıvely In each dıagram the percentage of heat passıng to the lıner pıston etc ıs plotted on a base of stroke bore ratıo A few notes of explanatıon are desırable

(1) In each case the total jacket heat ıs assumed to be 25% of the total heat supplıed

(2) The exhaust port loss is regarded as being independent of the stroke bore ratio

(3) The liner loss approaches zero as the stroke bore ratio approaches zero

(4) The cover plate loss and the piston loss are assumed to approach equality as the stroke bore ratio approaches zero

(5) The cover plate loss and the piston loss tend to 0 as the stroke bore ratio tends to ∞

(6) For finite values of the stroke bore ratio the trends of the curves have been guided by the data given above combined with a fairing up process

No great accuracy can be claimed for these diagrams but they have been checked from time to time against published data relating mainly to marine engines and the order of accuracy for individual items is probably round about $\pm 20\%$

The Heat Flux through the Walls —The foregoing data enable the heat flux through the walls of the liner cover plate and piston to be calculated approximately in any given case The mean flux through the liner is found by dividing the liner heat in B T U per hour by the area of the water cooled surface in sq ft In the cylinder for which Fig 123 has been drawn the cooled length of the liner is equal to the stroke plus about 80 % of the bore and the maximum flux is 2 5 times the mean In other cases the proportion of cooled length may be different but this is hardly likely to have any appreciable effect on the maximum flux Accordingly we shall assume that in all cases the maximum flux is 2 5 times the mean flux when the latter is calculated on an area equal to the product of the cylinder circumference $\times$ the stroke plus 0 8 times the bore

Let I = Indicated power per sq ft of piston area
B = Bore in feet
S = Stroke in feet

Consumption of fuel (18 000 B T U per lb) = 0 3 lb per I H P hr

l = Fraction of total heat supplied which passes through the liner (see Figs 124 and 125)

F = Maximum flux in 1000 B T U /ft ^{2} hr

Then —

$$F = \frac{\dfrac{I\pi B^2}{4} \times \dfrac{0\ 3 \times 18\ 000}{1000} \times 2\ 5l}{\pi B(S + 0\ 8\ B)} = \frac{3\ 4Il}{\left(\dfrac{S}{B} + 0\ 8\right)} \qquad (1)$$

The heat received by the cylinder cover plate cannot be quite equally distributed but must be somewhat more intense at the centre thermocouple readings given by Chalorer shewed a difference of only about 4% between the flux at the centre of the cover and that at a point situated at a distance of about three quarters of the cylinder radius from the centre Also some of the heat received by the underside of the plate is conducted away radially beyond the cylinder radius before reaching the water jacket The heat received by the plate divided by the area of the latter must therefore be an over estimate of the mean flux though it is probably a good approximation to the maximum flux Similar considerations apply to the piston We therefore estimate the maximum flux to the cover plate and to the piston respectively by dividing the heat received in each case by the piston area

Let c = Fraction of total heat supplied which passes through the cover plate

p = Fraction of total heat supplied which passes through the piston crown

(See Figs 124 and 125)

Then —

$$F = I \times \frac{0\ 3 \times 18\ 000}{1000}\ c\ (\text{or } p) = 5\ 4\ I\ c\ (\text{or } p) \qquad (2)$$

The values of c (or p) decrease as the stroke bore ratio increases

Examples —

	I	$\frac{S}{B}$	$\frac{3\ 4}{\frac{S}{B}+0\ 8}$	1	D		F [illegible]	F [illegible]	F [illegible]
4 Stroke Submarine Engine	170	1 0	1 88	085	055	065	2	51	60
4 Marine	75	1 5	1 48	115	040	050	13	16	20
4 Marine	85	2 0	1 21	13	030	040	1?	12	18
2 Naval	240	1 2	1 70	085	060	060	35	8	78
2 Marine	140	1 5	1 48	100	055	055	21	41	41
2 Stationary	160	2 0	1 21	120	045	045	23	39	39

These figures give an approximate guide to the values of the heat flux to be reckoned with in typical cases

Local Temperature Stresses —A knowledge of the heat flux enables the local temperature difference and consequently the local temperature stress to be estimated

Let t = Thickness of wall in inches

$(T_1 - T_2)$ = Temperature drop through the wall in deg F

k = Conductivity of cast iron in ft B T U hr units = about 25

E = Modulus of elasticity of cast iron = about 12×10^6 lb /in

λ = Coefficient of expansion of cast iron = 6×10^{-6}

Then the maximum local difference of temperature from the mean $= \frac{1}{2}(T_1 - T_2)$

$$= \frac{1000\ F}{2k} \times \frac{t}{12} = \frac{500\ Ft}{25 \times 12} = 1\ 67\ Ft$$

And the temperature stress f_t is given by —

$$f^t = \tfrac{1}{2}(T_1 - T_2)\ E\ \lambda = 1\ 67\ Ft \times 10^6 \times 12 \times 10^{-6} \times 6 = 120\ Ft \qquad (3)$$

The important point is that for a given value of the heat flux F the temperature stress is proportional to the wall thickness Consider for instance the two stroke Naval engine referred to in the above table For the cover and piston F = 78 and the local temperature stress = 120 × 78 = 9 360 lb / in per inch of thickness It is evident therefore that with ordinary materials the available range of safe thicknesses is limited whatever the bore of the cylinder may be The problem of the large highly rated cylinder is therefore reduced to finding methods of construction whereby comparatively thin walls may be used with safety The question as to what temperature stress can be considered safe is naturally an important one and a very difficult one to decide Most heat failures of covers and pistons are traceable to extensive temperature differences giving rise to stresses which are very difficult to estimate Cylinder liners and plain ribless covers and pistons afford better criteria and calculations of known designs on the lines indicated above indicate that the safe limit of temperature stress lies somewhere between 10 000 and 15 000 lb per sq inch The ultimate tensile strength of good cylinder iron is about 30 000 lb per sq inch and a safe limit of half this value viz 15 000 lb /in ^{2} may be suggested

The whole business is greatly complicated by the peculiar behaviour of cast iron at high temperatures According to the theory outlined above the hot side of the wall should be in a state of compression and the cold side in tension Actually cracking when it occurs almost always takes place on the hot side leaving when cold a gaping fissure which tends to close

up during running of the engine From this fact it appears that the temperature stresses are annealed out during prolonged running by rearrangement of the particles and that cracking takes place on cooling by the reappearance of the temperature stresses with reversed sign Nevertheless it is to be expected (apart from the phenomenon of growth) that the stresses induced during cooling will be proportional if not actually equal to those induced by the initial heating before annealing has taken place

Local Temperature Stresses in Pistons and Covers —It is interesting to see what results are yielded by formula (3) in representative cases

(1) Four stroke engine 12 ×18 I=75 t=$1\frac{1}{8}$ for cover
$f_t = 120 \times 20 \times 1·125 = 2700$ lb /in
Local temperature stress obviously negligible

(2) Four stroke engine 30 ×45 I=85 t=2 for cover
$$f_t = 120 \times 20 \times \frac{85}{75} \times 2 = 5500 \text{ lb /in}^2$$

(3) Two stroke engine 30 ×45 I=140 t=$1\frac{1}{2}$ for cover
$f_t = 120 \times 41 \times 1·5 = 7400$ lb /in ^{2}

(4) Two stroke engine 20 ×24 I=240 t=0·875 for cover
$f_t = 120 \times 78 \times 0·875 = 8160$ lb /in ^{2}

(5) Four stroke engine 30 ×45 I=85 t=$1\frac{3}{4}$ for piston
$$f_t = 120 \times 16 \times \frac{85}{75} \times 1·75 = 3800 \text{ lb /in}$$

(6) Two stroke engine 30 ×45 I=140 t=$1\frac{3}{4}$ for piston
$f_t = 120 \times 41 \times 1·75 = 8600$ lb /in ^{2}

In the above all pressure stresses and extensive temperature stresses are left out of consideration These figures shew that if undue thicknesses of metal are avoided the local temperature stresses can be kept below a moderate limit

Local Temperature Stresses in the Liner —In the case of the liner the additional stress due to pressure is easily allowed for Assuming a maximum working pressure of 500 lb /in ^{2} the pressure stress f_p is given by —

$$f_p = 250 \text{ B/t} \qquad (4)$$

Where B=cylinder bore and t=liner thickness in inches The combined pressure and temperature stress f_c is the sum of f_p as given above and f as given by (3) For example in a

four stroke engine of 24" bore × 36" stroke with I = 75 we have (since F = 13, see table p 173) —

$$f = (250 \times 24/t) + (120 \times 13 \times t)$$
$$= 6000/t + 1560\,t \qquad (5)$$

Values of f_p, f_t and f_c are plotted on a basis of t in Fig 126 and f has a minimum value of 6100 lb/in.² when t = 2 and

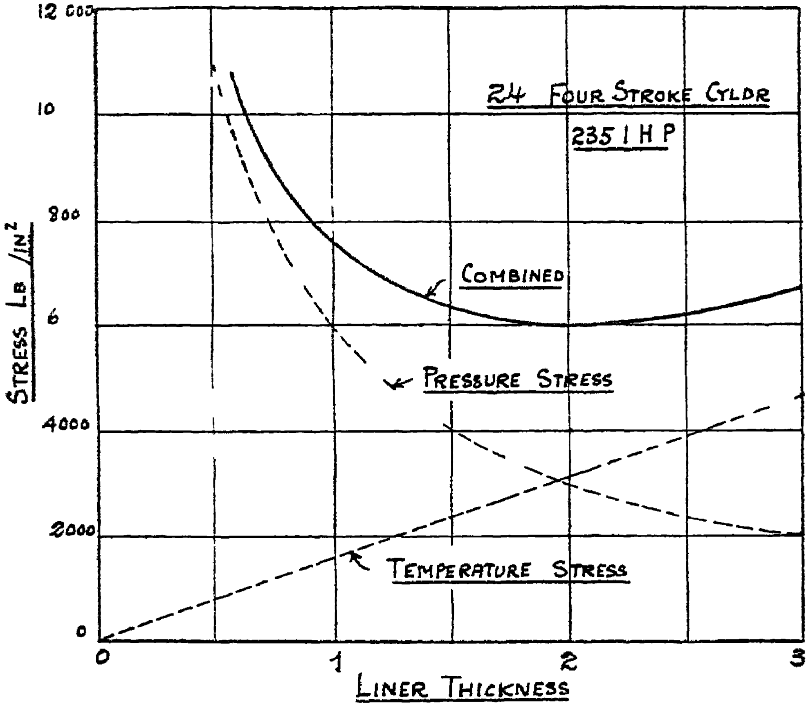

Fig 126

this value occurs when $f_t = f_p$. A similar diagram is shewn in Fig 127 for a two stroke engine 17.5 bore 26 stroke with I = 140 developing the same I H P as the four stroke engine. In this case the minimum value of f_c is higher viz 6600 lb/in.² and the optimum thickness is about $1\frac{3}{8}$.

Both these values of the thickness are in good agreement with average practice with reference to the thickness under the top flange of the liner

The following features of these diagrams may be noticed

(a) There is a certain thickness of the liner for which the combined stress is a minimum

(b) The minimum stress is slightly higher for the two stroke cylinder than for the four stroke cylinder of the same I H P at the specified rating

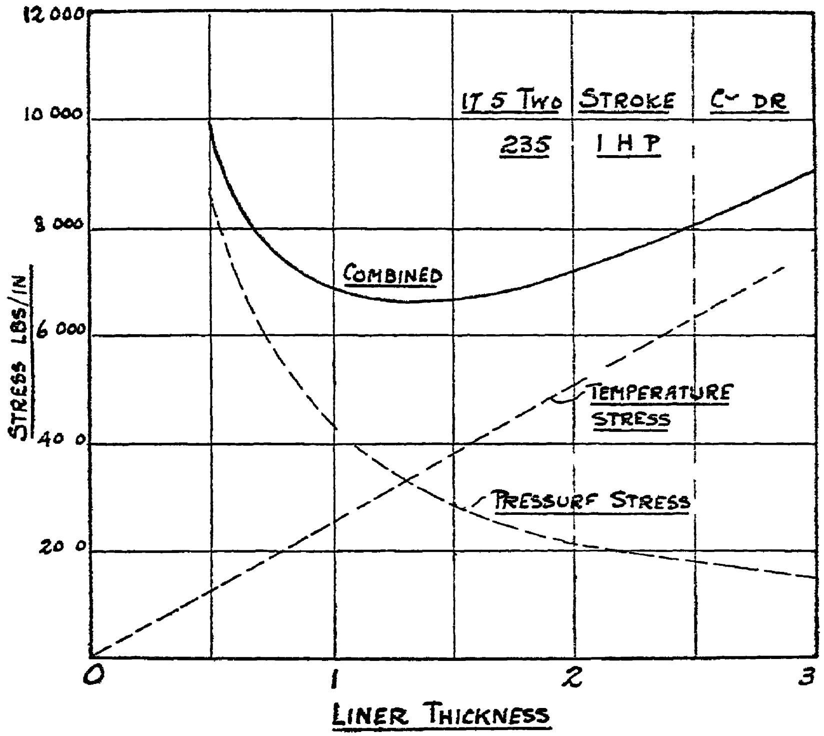

FIG 12

(c) The minimum stress increases as the cylinder size is increased

(d) Considerable variations from the most favourable liner thickness are possible with small influence on the combined stress

The last consideration is interesting in view of the thickening of the liner at the flange end which occurs in most designs An increase of about 30% increases the stress according to the diagram by about 4 or 5% If as in some designs the thickness

at the flanged end is about double the normal thickness the increase of stress may be very considerable in highly rated engines This point is illustrated by Fig 128 which has been drawn up for a 30 ×45 two stroke engine rated at I=166 In this case —

$$\begin{aligned} f_c &= (250 \times 30/t) + (120 \times 25\ 5\ t) \\ &= 7500/t + 3050\ t \qquad (6) \end{aligned}$$

The minimum combined stress is 9500 lb /in ^{2} with a thickness of about $1\frac{9}{16}$ The more usual thickness of about $2\frac{1}{4}$

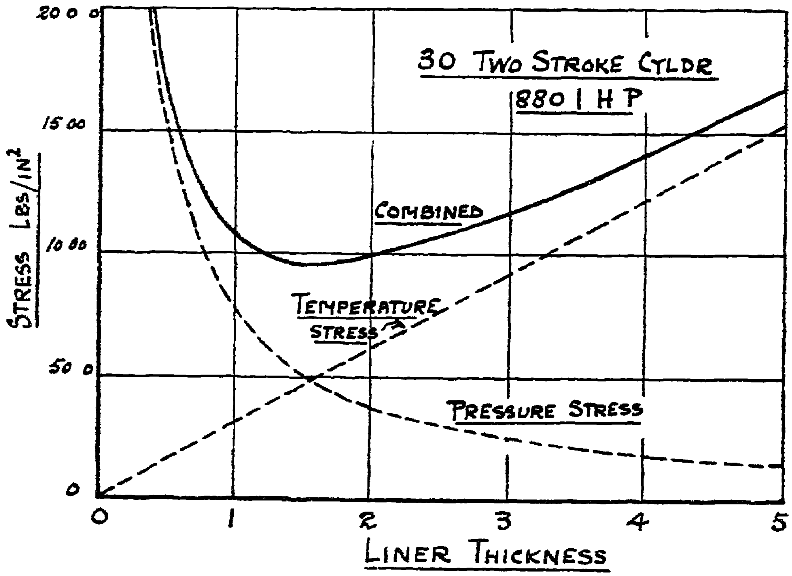

Fig 128

increases the stress to about 10 500 lb /in and a 4^1 flange would involve a stress of over 15 000 lb /in ^{2} This suggests the advisability of avoiding such heavy flanges in large cylinders subject to a high flux and experience confirms this view

Extensive Temperature Stresses —Extensive temperature differences giving rise to high stress arise when the liner and jacket are cast in one piece The jacket portion remains nearly stone cold whilst the liner acquires a certain mean temperature which under a constant load slowly rises with

time on account of deposits of scale until such time as the latter be removed so that a construction which may be safe when new tends to become unsafe in service unless deposits are prevented Numerous failures of gas engine cylinders with integral jackets and liners have deterred most Diesel Engine builders from using this construction except in the very smallest sizes

The traditional type of circular cylinder cover is liable to temperature stresses arising from the rigid connection of the hot bottom plate to the cool surrounding walls Anything which tends to increase the mean temperature of the bottom plate tends to increase the temperature stress Failures are therefore most frequent with highly rated engines using hard water Tendency to failure is minimised by —

(1) Conservative rating
(2) Use of soft water
(3) Introducing the water at high velocity at the centre of the cover
(4) Using a low outlet temperature

In this respect marine engines are at an advantage in working for the greater part of the time under conditions which admit of effective cooling with a liberal supply of clean water

With two stroke engines at normal ratings these extreme temperature differences are greatly accentuated and special constructions admitting of greater freedom for expansion become necessary Some of these have already been referred to in Chapter VIII together with some constructions adopted in the larger sizes of four stroke engines

Uncooled Pistons —With water or oil cooled pistons the temperature of the crown is probably fairly uniform over the greater part of the area with a gradual falling off at the edge and down the sides With uncooled pistons on the other hand there is a steep temperature gradient towards the edge of the crown due to the conduction of the heat to the liner The simplest ideal case for calculation purposes is that of a flat crown of diameter B (feet) and small thickness d (feet) subject to a uniform flux F (1000 B T U /ft hr) If r is any radius less than B/2 the flow received inside this radius is

$$\pi r^2 F \times 1000 \text{ B T U per hour}$$

and this must be equal to the temperature gradient—dT/dr ×

conductivity k and the sectional area $2\pi r d$ we have therefore —

$$\frac{dT}{dr}=\frac{-\pi r^2F\times 1000}{2\pi r dk}=\frac{-rF\times 1000}{2dk}$$

$$T=Tc-\frac{r^2F\times 1000}{4dk}$$

$$\text{and } Tc-Te=\frac{B^2F\times 1000}{16dk} \qquad (7)$$

where Tc = Temperature at the centre
and Te = edge

Converting to inch units (7) becomes —

$$Tc-Te=\frac{B^2F\times 1000}{16\times 12d\times 25}=0\ 208\ B^2F/d \qquad (8)$$

Under the simple conditions postulated the stress due to the uneven distribution of temperature must be proportional to (Tc—Te) and temperature measurements made in cases where experience indicates that cracking is not to be anticipated indicate that a safe value of Tc—Te is about 400 F

Substituting this value in (8) we obtain the relation —

$$d=\frac{0\ 208\ B^2F}{400}=\frac{B^2F}{1920} \qquad (9)$$

The thicknesses of crown which result from this relation for two values of F viz 16 (four stroke engine rated at I=75) and 41 (two stroke engine rated at I=140) are tabulated under —

Bore in ins	5	10	15	20	25
d (F=16)	21	84	1 87	3 35	5 20
d (F=41)	53	2 15	4 80	8 50	13 40

The small thickness (0 21) required by the 5 piston is characteristic of motor car practice and the other figures shew the hopelessness of trying to emulate the slender proportions of petrol engine practice in the design of medium and large size uncooled pistons for internal combustion engines The point to which attention is here drawn has sometimes been overlooked by critics of oil engine and gas engine designs

The table also shews the limitations of uncooled pistons subject to the assumed values of the flux It is clear for example that at the higher rating (F=41) uncooled pistons

of cast iron are impracticable for cylinders above about 12″ diameter or so

Assuming that the temperature difference is confined to some limit such as that suggested the safety against cracking will depend on the form of the crown whether concave or convex the degree of curvature the connection to the skirt etc and the physical and chemical properties of the material in relation to high temperatures If the temperature difference is constant in different sizes the maximum temperature will be greatest (other things being equal) in the largest size on account of the increased length of the paths of heat flow from the edge of the crown to the jacket It can hardly be doubted that too high a maximum temperature tends towards cracking on account of the material becoming locally plastic when hot and developing high tensile stress when cooled

The considerations advanced in this section do not enable the temperature stresses to be calculated but they throw some light on necessary limitations and afford a basis of comparison on which proposed designs may be compared with known successful ones of similar form but of different size and differently rated

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CHAPTER X

RUNNING GEAR

Trunk Pistons —These are so well known in connection with petrol motors gas engines and the like that a general description is unnecessary and we may at once proceed to the consideration of their special requirements in Diesel Engine construction The difficulties involved in combining the piston proper with the crosshead arise chiefly from the heat which reaches the gudgeon pin bearing by conduction and radiation and the high pressures dealt with in Diesel Engines (as compared with gas engines) necessitate a high specific pressure at this bearing owing to the limited space available The most serious troubles to be anticipated are piston seizures which like all other heat troubles are more pronounced in large than in small engines For these reasons uncooled trunk pistons are seldom used for cylinders exceeding about 22 in in diameter For marine engines the inaccessibility of the gudgeon pin bearing is often considered to be a serious objection for all but the very smallest cylinders It is possible that prejudice has a little to do with this view and it is interesting to note that Diesel Marine Engines of fairly large size (1000 H P for example) apparently give good service with trunk pistons It is evident that a really efficient system of oil or water cooling is essential in large sizes in order to conduct away the heat received by the crown without the latter attaining too high a temperature and cracking in consequence

Oil cooling affords the simplest solution as the oil is readily supplied from the forced lubrication system and leakage does no harm

Water cooling by means of telescopic pipes is also used in these cases the water is squirted into the piston jacket in the form of a jet and concentric guard tubes are arranged to catch the spray and the outlet water

Material —The use of cast iron for pistons is almost universal on account of its good wearing properties and its cheapness Owing to the low guide pressure the quality of the metal is

probably of minor importance so far as wear is concerned as the latter is in any case hardly measurable On the other hand so far as that portion of the piston is concerned which is in contact with the working fluid (viz the piston crown) the quality of the metal is of great importance in determining the liability or otherwise of the crown to crack under the influence of heat According to Mr P H Smith the irons which give the best results are those of the coarsest possible grain

Experience in other directions seems to indicate that a large carbon content and low percentage of phosphorus are favourable

At the first glance it might appear strange in view of the

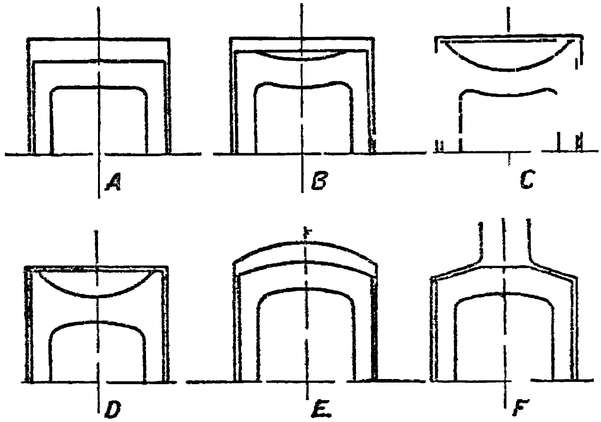

FIG 129

fact that heating produces compression at the point of maximum temperature that the cracks start at the centre of the crown on the side in contact with the gases but the facts are easily explained by some such hypothesis as the following —

The local heating causes local compressive stress of high intensity which in course of time causes the particles to rearrange themselves in such a manner that this stress is reduced On cooling the contraction of the surrounding metal induces tensile stresses in the centre equal in amount to the extent by which the original compressive stress has been reduced Considerable support is afforded to this theory by the fact that the cracks develop in the course of time into fissures shewing that the material has contracted circumferentially Also the plastic

deformation of cast iron at a red heat is often observed in such familiar articles as kitchen ranges and the like in which no special provision is made for expansion or growth

Shape of Piston Crown —Fig 129 shews some alternative shapes

Of these types C is the best for combustion though types A and B are little inferior in this respect With existing methods of fuel injection type E is quite inadmissible on the score of combustion unless the cylinder cover is concave downwards and even then its efficiency is very doubtful On the other hand type F with the shape of combustion space indicated appears to give good results due doubtless to the manner in which the charge of air is concentrated A little reflection shews that the curved shape of type B gives rise to greater compressive stress at the centre on the combustion side than type A on account of the bending action which arises when a state of temperature stress comes into existence Type D is very liable to failure unless a spreading flame plate is used From this it would appear that of all the types illustrated B and D are the most liable to failure In practice types B and C are the most usual and their inherent weakness in the larger sizes (small pistons rarely crack) is guarded against by

(1) Careful selection of material
(2) Providing a considerable thickness of metal
(3) Oil or water cooling
(4) Providing loose crowns or central cores

The provision of a great thickness of metal assists matters in two ways —

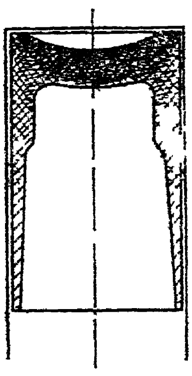

FIG 130

(a) By reducing the bending stresses due to both temperature (probably the most vital) and pressure
(b) By giving additional area for heat flow to the walls of the liner

Fig 130 is an attempt to give a diagrammatic representation of the heat flow by increasing the intensity of shading towards the parts having the higher temperatures The usual form of water cooled

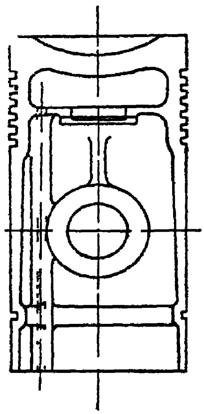

FIG 131

trunk pistons is shewn in Fig 131 and the means adopted to convey the water to and from the water space will be discussed later In the earlier types the space was intended to be full of water but in more recent designs the tendency is to rely on a small flow of water part of which is evaporated and absorbs a relatively large quantity of heat in latent form

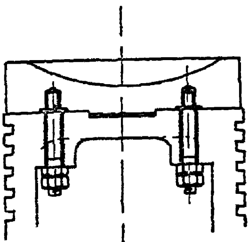

Fig 132

Fig 132 shews a loose piston top secured by four studs the holes for which have sufficient clearance to allow for the expansion of the former Fracture of such a loose top due to temperature effects is improbable and if it occurs the cost of renewal is trivial

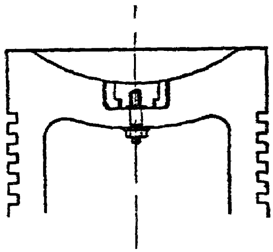

Fig 133

Fig 133 shews a similar design patented by Mr P H Smith and applied by him to pistons in which cracks had already appeared

Proportions of Trunk Piston Bodies —The usual proportions found in practice are discussed below with reference to Fig 134 which is purely diagrammatic

The thickness (C) of the crown (with uncooled pistons) increases rapidly as the bore is increased for reasons which have been noticed above and the following figures are a guide to good practice —

Bore of cylinder	Thickness C
10	$1\frac{1}{4}$
12	$1\frac{5}{8}$
14	$2\frac{1}{4}$
16	3
18	4
20	5

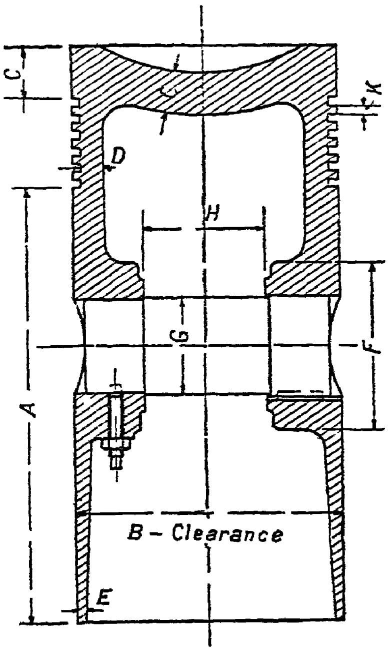

Fig 134.

The distance from the top of the piston to the first ring may conveniently be made equal to C With

the above values of C this prevents the first ring being placed in too high a position where its proximity to the source of heat would cause it to become stuck with carbonised lubricating oil in a short time The rings themselves may be of square section with R =0 025 to 0 033 B The gap between the ends of the ring when free may be about 2 50 times R [1] The number of rings fitted varies from about five in small to eight in large cylinders and the space between consecutive rings is not as a rule less than R The construction of piston rings is rapidly becoming a specialised branch of industry and details will not be given here A method of designing and manufacturing piston rings is described by Guldner in considerable detail (Design and Construction of Internal Combustion Engines) It is important to prevent the joints of the rings working into line during service and a method of fixing them is shewn in Fig 135

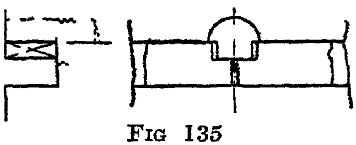

Fig 135

Dimension A varies greatly in different cases The maximum value found in practice viz A =2 B gives a piston of ideal running properties but is seldom fitted nowadays owing to considerations of first cost A =1 4 B to 1 6 B represents good average practice Still smaller values of A are sometimes used but are not to be recommended The provision of a long piston skirt is advantageous from the following points of view —

(1) Rendering possible a low and therefore comparatively cool location for the gudgeon pin bearing
(2) It minimises piston knocks
(3) It facilitates the flow of heat away from the crown by providing a large surface in contact with the cylinder walls

The upper part of the piston body is turned taper to allow for expansion and the approximate allowances to be made on the diameter are given in Fig 136 in which the taper is greatly exaggerated The diameters of the piston ring grooves are figured by allowing all the grooves to have the same depth below the tapered surface The clearance behind the grooves

[1] R denotes the thickness of the ring measured radially The bars K between the rings may be equal to or slightly greater than R

should be a practicable minimum The gudgeon pin is usually located either at the centre or slightly above the centre of the parallel part of the body and allowance is made for possible local distortion due to driving in the gudgeon pin or to expansion of the latter by relieving the piston surface to the

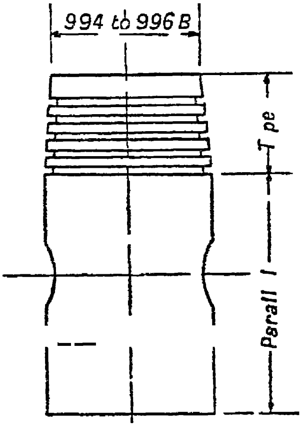

Fig 136

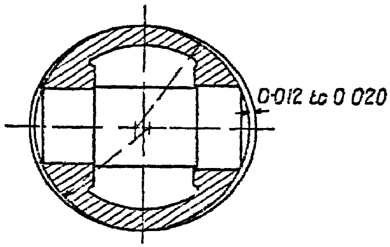

Fig 1

extent of about 20 thousandths of an inch in the manner indicated in Fig 137

Approximate figures for other main proportions are given below —

C (see Table above)	F =2 G
D =0 07 B	G =0 4 B
E =0 033 B	H =0 5 B

Gudgeon Pins —Alternative forms of gudgeon pins are shewn in Fig 138 type A being most generally used The pin itself is of special steel case hardened and ground if working in a bronze bearing If the bearing is white metalled the case hardening is unnecessary The pin is a driving fit in the body and is secured in the manner indicated in the illustrations Lubrication of the pin may be effected in various ways —

(1) Forced lubrication by means of a pipe or drilled hole leading from the big end of the rod

(2) By means of a groove or pocket on the surface of the piston communicating with the surface of the gudgeon pin and fed by means of a fitting similar to that shewn in Fig 105 (see Fig 139),

Assuming the proportions given in the previous article the maximum bearing pressure works out to about —

$$\frac{0.785 \times 500}{0.4 \times 0.5} = \text{about } 2000 \text{ lb per sq in}$$

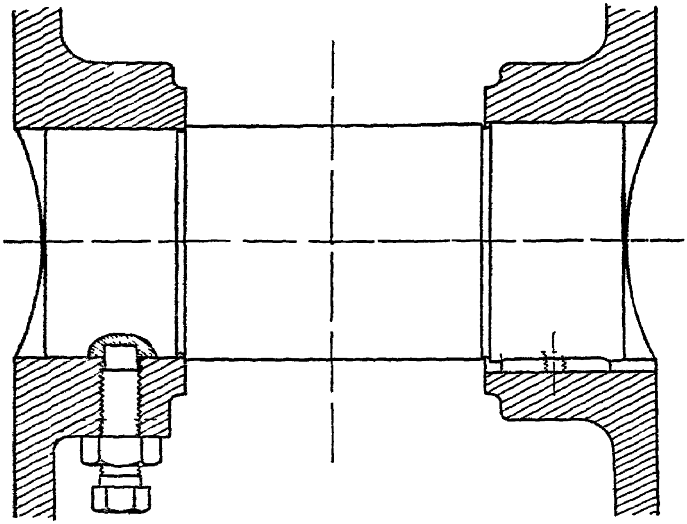

FIG 138 Type A

and it is hardly surprising that good bronze appears to be preferable to white metal for this bearing

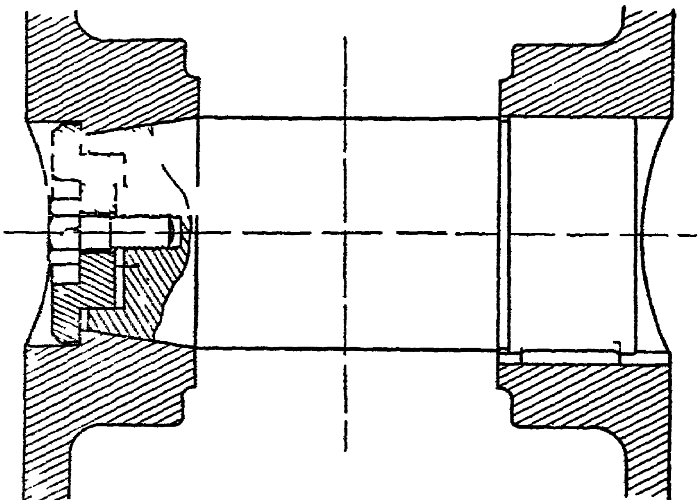

FIG 138 Type B

Miscellaneous Points of Detail —Where forced lubrication is employed it is important to prevent oil from splashing on to the hot piston crown and some arrangement of baffles is very desirable This may take the form of a plate or diaphragm across the piston body A piston ring at the lower end of the piston is useful in removing superfluous oil from the liner and in effecting a good distribution of the film

It is generally necessary to cast two shallow recesses in the

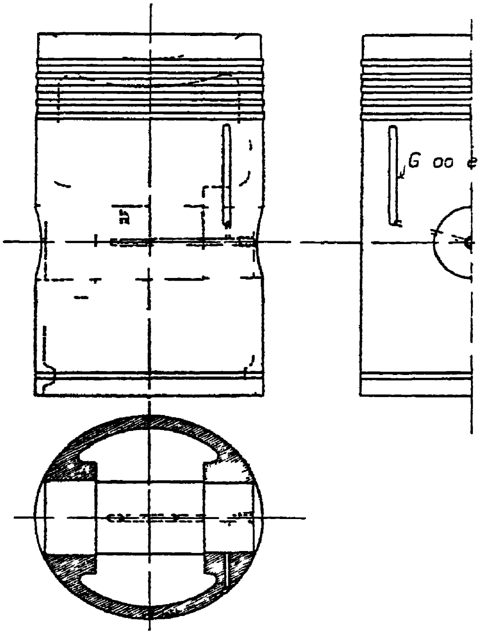

FIG 139

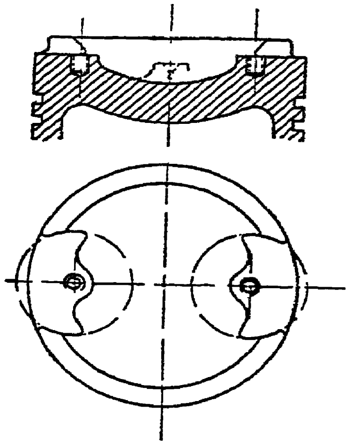

FIG 140

piston crown to clear the air and exhaust valve heads at the top dead centre and the positions of these being remote from the point of greatest temperature are usually chosen for two tapped holes to receive lifting bolts Holes should not be drilled in the centre of the crown and if a turning centre is necessary a special boss should be cast for this purpose and turned off afterwards (see Fig 140)

So far nothing has been said about pressure stresses and the bearing pressure on the piston body considered as a crosshead as these appear to be irrelevant Temperature considerations determine the proportions of the piston body and the gudgeon

bearing is made as large as the limited space allows The guide pressure between the piston body and the liner works out at a very moderate figure and measurements of pistons after long periods of service fail to disclose any appreciable wear and justify the conclusion that under normal running conditions the piston body floats on a film of oil In cases of seizure the cause of abrasion is usually traced to local distor

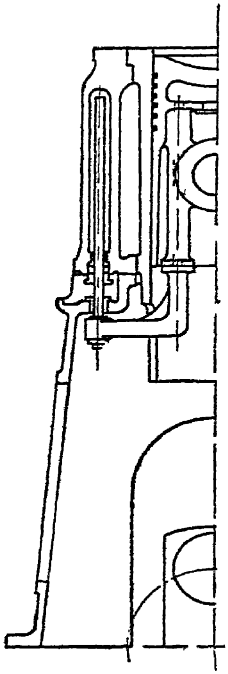

Fig 141 Type A

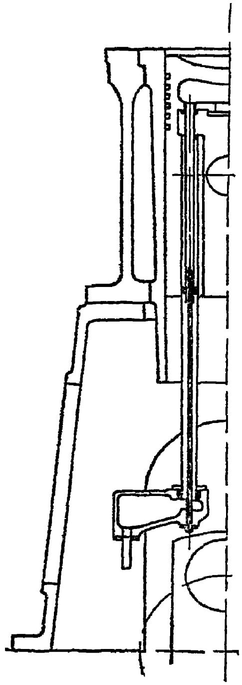

Fig 141 Type B

tion sometimes assisted by the destruction of the oil film by the presence of viscous deposits

Water Cooling —Two systems of conveying the water to and from the piston are shewn in Fig 141 In both systems the aim is to render the success of the scheme independent of the water tightness of the various joints involved In type B inaccuracies of alignment are allowed for by a ball joint at the foot of the stationary tube

Pistons for Four Stroke Crosshead Engines —These are generally made of not much greater length than is necessary to accommodate the rings eight to ten in number The pro

vision of an extra number of rings above what is considered sufficient for a trunk piston may be attributed to —

(1) The throttling effect lost by discarding the piston skirt
(2) The lower speed of revolution usually associated with engines of the crosshead type

Cooling by means of a blast of air has been used (apparently successfully) in cylinders of medium size but water cooling is now almost universal Fig 142 exhibits different forms of piston having one feature in common viz a self supporting crown There seems to be no doubt that internal ribs as

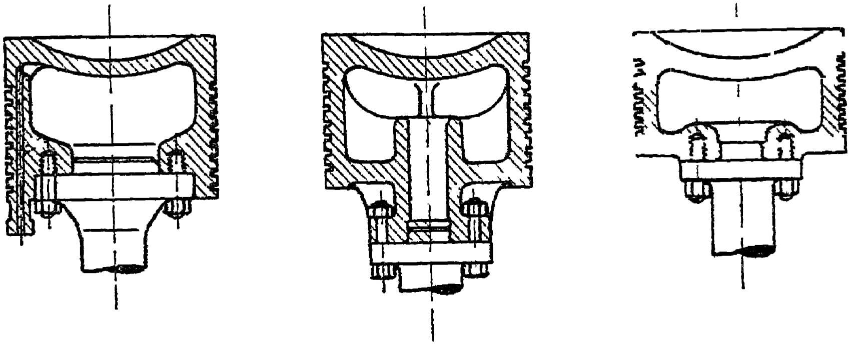

Fig 142

used in the past are conducive to cracking both of cooled and uncooled pistons They are therefore omitted in most modern designs

Two systems of admitting the water to and leading it away from the piston are shewn in Fig 143 which is practically self explanatory In each case there is a tray or diaphragm separating the crank case from the cylinder so that any small leakage cannot reach the former

Pistons for Two Stroke Crosshead Engines —The existence of ports in communication with the exhaust pipe at the lower end of a two stroke cylinder necessitates the provision of a skirt or extension of the piston to prevent the uncovering of these ports when the piston is at the top dead centre The skirt usually takes the form of a light drum secured to the piston by a number of well locked studs It is common practice to arrange one or two inwardly expanding rings at the lower end of the cylinder to prevent leakage past the skirt (see Fig 144)

If the cylinder liner is of sufficient length these exhaust rings may be located in the skirt itself as in Fig 145 Such an arrangement involves a higher engine than that of Fig 144 but facilitates conduction of heat from the piston and incidentally secures a lower mean temperature for the cylinder liner The construction of the piston proper is generally similar to that of a four cycle engine but it must be borne in mind that the conditions as to temperature are more severe than in a four stroke engine of the same size working at the same mean indicated pressure so that the remarks of the preceding article in reference to ribs under the crown apply with still greater force to two stroke engines

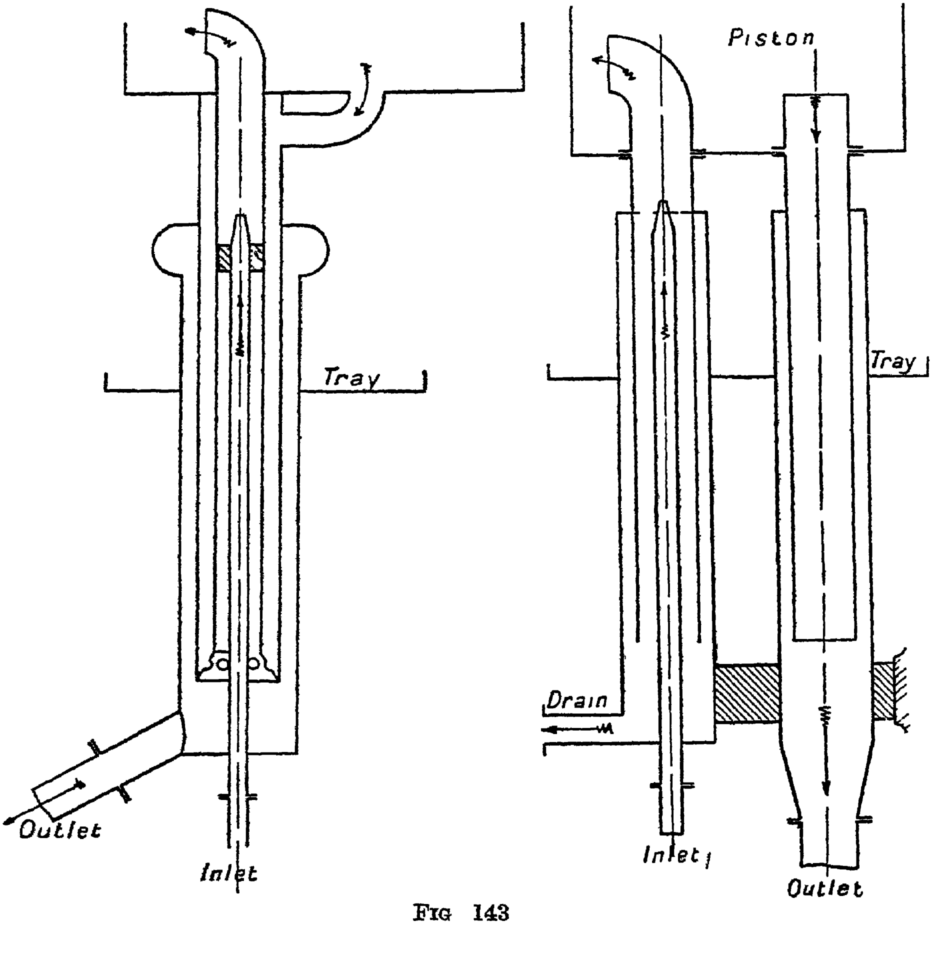

FIG 143

Some of the recent two stroke engines of various makers have been fitted with pistons in which the crown is free from ribs and the skirt is extended to do duty for the piston rod as shewn more or less diagrammatically in Fig 146

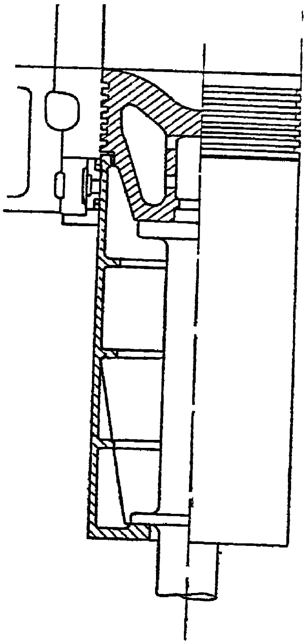

Fig 144

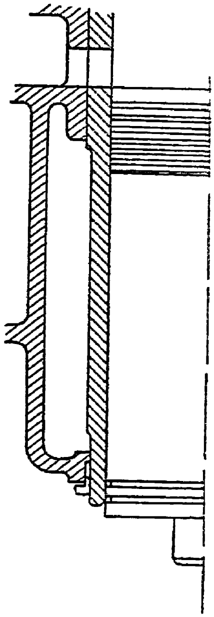

Fig 14

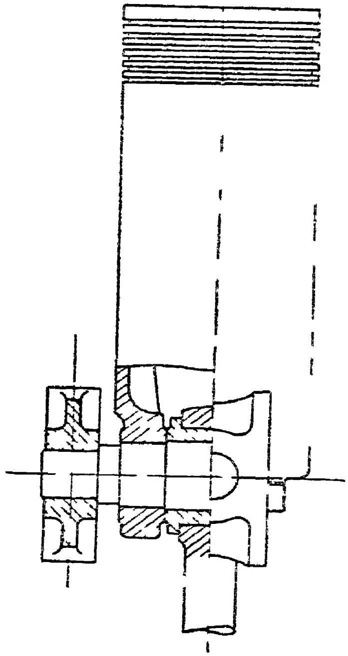

Fig 14

Piston Rods —The forces to which a piston rod is subject are —

(1) The pressure load of the piston which attains a maximum of about 500 lb per sq in of piston area with an occasional explosive load of about double this intensity at the top dead centre

(2) A load due to the inertia of the piston (including the water therein) and the rod itself This also attains its maximum at the top dead centre but is opposite in direction to the pressure load The maximum intensity of the inertia load seldom exceeds 70 lb per sq in of piston area in commercial engines

(3) Friction of the piston rings on the liner This effect has its maximum at the top firing centre and acts in the

same direction as the inertia so far as the expansion stroke is concerned It has been proved that piston ring friction absorbs about 5% of the indicated power and assuming (as seems probable) that the friction is at every point proportional to the cylinder pressure it appears that the maximum friction is equivalent to about 10 lb per sq in of piston area

(4) Fluid friction due to shearing of the lubricating oil film

The joint effect of these forces amounts then to about 420 lb compression at firing dead centre and about 70 lb tensile at the end of the exhaust stroke per sq in of piston area

Unfortunately the fatigue stress of steel between limits of this sort appears not to have been determined yet but the value is probably in the neighbourhood of 35 000 lb per sq in for 30 ton steel A table of buckling loads for circular mild steel rods with rounded ends is given below from which it will be seen that the question of buckling appears to be irrelevant in view of the fact that the ratio of length to diameter of rod does not usually exceed 15 and is usually less In this case and in cases of connecting rods also the use of a strut formula such as Euler s in conjunction with a large factor of safety would appear to be irrational (see page 205)

L—D =Length—Diameter	5	10	15	20	30
Buckling stress lb /sq in	63 000	56 000	45 000	33 000	21 000
L—D =Length—Diameter	40	50	80	100	
Buckling st ess lb /sq in	13 000	9 000	4 000	2 500	

In current practice the diameter of the piston rod is made from 0 23 to 0 3 of the cylinder bore corresponding to a maximum compressive stress under normal working conditions of about $\frac{420}{(0\ 23)^2}$=8000 lb /sq in to $\frac{420}{(0\ 30)^2}$=4700 lb /sq in The higher of these figures corresponds to a factor of safety under fatigue conditions of about 35 000—8000=4 35 which appears ample The stress under an explosion of 1000 lb per sq in would be $\frac{1000}{(0\ 23)^2}$=19 000 lb /sq in and the factor of safety against buckling (assuming $\frac{L}{D}=15)=\frac{45\ 000}{19\ 000}=2\ 35$ which would appear to be sufficient in view of the provision of a relief valve Nevertheless the lower stresses are generally preferred particularly in marine practice

The upper end of the rod usually ends in a circular flange for carrying the piston to which it is secured by a row of studs proportioned to the inertia load of the piston with a very moderate stress allowance in order to give a good margin for dealing with such emergencies as seized pistons The lower end is sometimes secured to the crosshead by means of a flange as in Fig 147 or in the manner indicated in Fig 148 In either case it is reasonable to make the connection of the same strength as that between the rod and the piston assuming that ample provision has been made for the contingencies referred to

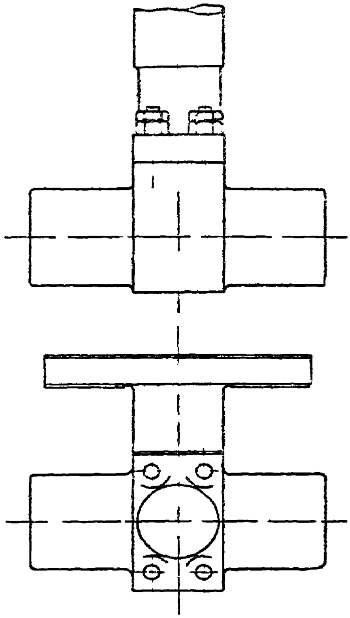

Fig 14

Crossheads and Guides — For marine engines the slipper guide shewn in Fig 149 is the favourite The bearing surface is usually made equal to the piston area [1] and the maximum bearing pressure with a con necting rod 4 5 cranks long then has a value of about 55 lb per sq in The slipper itself is of cast steel white metalled on ahead and astern faces The studs securing the slipper to the gudgeon block must be adequate to carry the maximum guide pressure when running astern The area of the gudgeon bearing is based on a bearing pressure of about 1500 lb per sq in The ahead guide face is of cast iron provided with water cooling The astern bars are frequently of forged steel secured by fitting bolts The stress in the latter is usually very moderate as stiffness is the chief con sideration The proportions given in Fig 149 are of course approximate only and subject to modification to suit different conditions

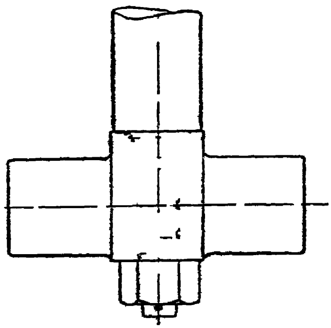

Fig 148

The type of guide block indicated on Fig 146 is well known

[1] *i e* the sectional area of the cylinder

in connection with paddle steamers and locomotives and needs no further description here For land engines double semicircular guides are sometimes used particularly when the cylinder and frame are cast in one piece In general the cross heads and guides used in Diesel Engine construction differ but

Width of crosshead face 0 75 B
Depth 1 13 B
Thickness plate 0 10 B

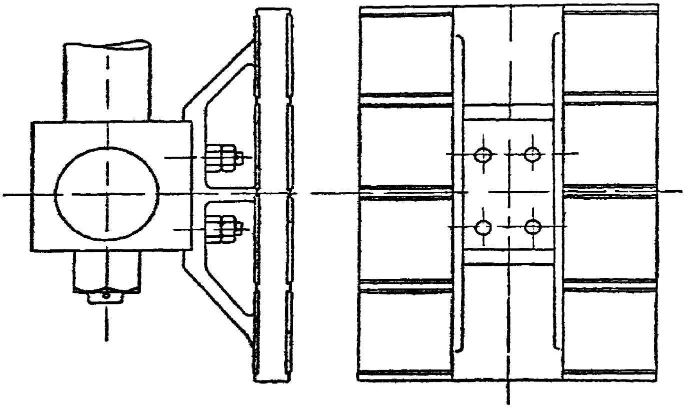

FIG 149

little from those commonly fitted to steam engines and large gas engines the most important point of difference being the gudgeon pin and its bearing which require to be liberally dimensioned to withstand the high maximum pressures to which they are subject It is also desirable to provide means to prevent carbonised oil from the cylinder from reaching the guide surface

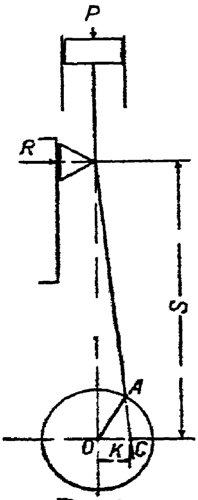

FIG 150

Guide Pressure Diagrams—A diagram shewing approximately the guide pressure at any crank angle is very simply obtained from the twisting moment curve in the manner described below with reference to Fig 150

P = Piston load in lb
R = Guide reaction in lb
T = Twisting moment in in lb
S = Height of gudgeon pin centre above the centre of the crank shaft

$$\text{Now } R = \frac{P\,k}{S}$$

$$\text{But } T = P\,l$$

$$\text{Therefore } R = \frac{T}{S}$$

The rule is therefore Divide the turning moment at any instant by the distance from the gudgeon pin centre to the crank shaft centre and the result is the guide reaction at the same instant It would appear that the guide reaction and the twisting moment should change sign simultaneously This is not quite the case for the following reason —

The twisting moment curve contains an inertia element in which an approximation is obtained by dividing the mass of the connecting rod in a certain proportion between the revolving and reciprocating parts This approximation though good so far as vertical forces are concerned gives very inaccurate values for horizontal forces Also the centrifugal effect of the revolving parts of the rod influences the guide reaction but not the twisting moment These discrepancies are of very small importance with the piston speeds at present obtaining For a full discussion of the influence of the connecting rod inertia forces on the guide reaction the reader is referred to Dalby s Balancing of Engines A table of values of $\frac{S}{l}$ where l = the length of the connecting rod is given below for various crank angles assuming a rod 4 5 cranks long

Crank angle	0	20	40	60	80	100	120	140	160	180
Values of $\frac{S}{l}$	1 22	1 21	1 16	1 09	1 02	0 94	0 87	0 82	0 79	0 78

Connecting Rods —The material for connecting rods is generally Siemens Martin steel of the same quality as that used for the crank shaft Stampings are sometimes used for small engines and if large quantities are made at a time this is an economical way of producing a rod of H section if machining all over is not considered essential Cast steel has been used on the Continent for gas engine connecting rods but the author has not met with this practice in Diesel Engine construction either British or continental

Connecting Rod Bodies —The section of the body or shaft of the rod is generally circular or part circular with flattened

sides The latter section is slightly lighter for a given strength but involves an extra machining operation For extreme lightness an H section of rod milled from the solid (like a locomotive rod) would appear to be advantageous The body usually tapers gently from the big to the small end and it will be shewn later that this practice is a rational one from considerations of strength It is interesting to note that the engineers of the North East Coast in their recent specification for standard reciprocating marine steam engines recommend parallel rods and it seems possible that the extra cost of

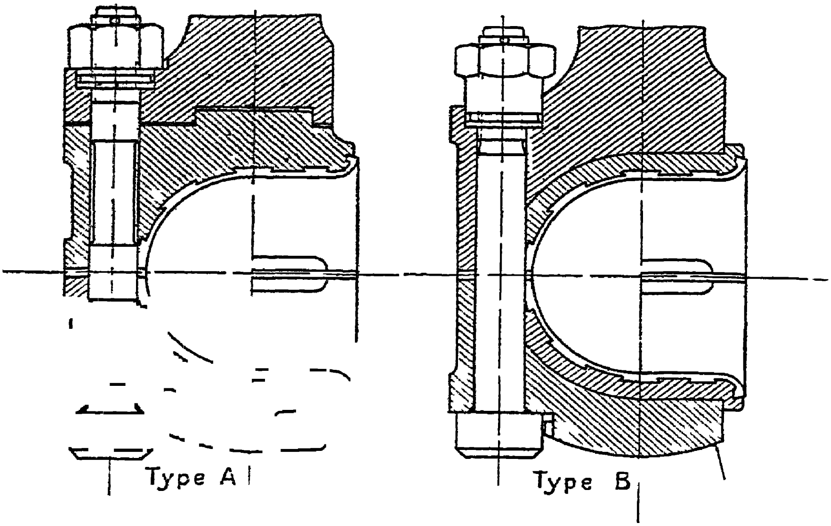

Fig 151

material involved is discounted by the reduced cost of forging and machining

Big Ends —Fig 151 shews two forms of big end type A being the cheapest and that most commonly used Type B is the strongest but suffers from the disadvantage of providing no facility for adjusting the compression by means of liners

Returning to type A the brasses are usually of cast steel lined with white metal With a stronger section as shewn in Fig 152 cast iron may be used instead of cast steel with satisfactory results but the practice is uncommon The bolts are frequently reduced to the core diameter between fitting lengths as shewn in Fig 151 but it appears that full diameter

bolts are stronger under the conditions to which they are subject in trunk piston engines In addition to tensile stresses the big end bolts have to resist shearing forces between the two brasses and also between the crown brass and the palm end of the rod Partial relief of this duty is afforded by the following means one or more of which are generally used in good designs —

(1) Spigoting the two brasses into each other as in Fig 153 This is very rarely done
(2) Spigoting the crown brass into the palm of the rod as in Fig 151 Type A
(3) Providing fitting rings half in the palm and half in the crown brass at the bolt holes (Fig 154)

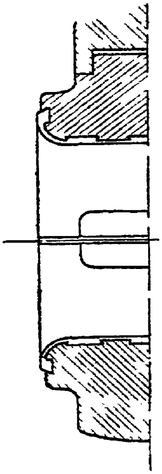

Fig 152

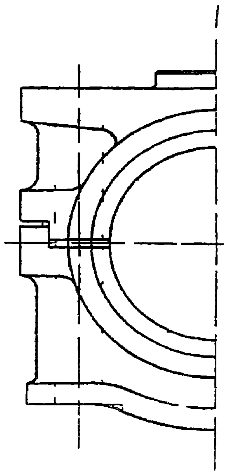

Fig 153

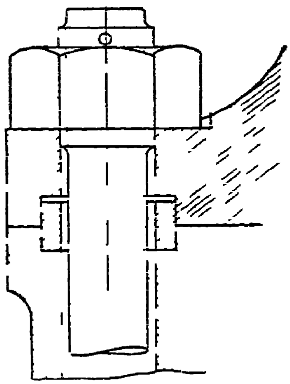

Fig 154

Small Ends —Various types of small end for trunk engine rods are shewn in Fig 155

With type A the chief difficulty is to find room for bolts of adequate strength If a big end of type B Fig 151 be fitted it becomes necessary to make provision at the small end for adjusting the compression as in Fig 155 types B and D Type C combines strength and adjustability of the bearing itself but makes no provision for altering the compression Type E contains a solid bush which must be replaced when worn and is therefore only suitable for small engines and adequate section of metal round the bush must be provided to prevent the hole becoming enlarged (see approximate

proportions on Fig 155) The brasses are usually of phosphor bronze

The forked end of a marine connecting rod is shewn in Fig 156 on which approximate proportions are noted in terms of the cylinder bore It differs from the similar member of a marine steam engine chiefly in the following points —

(1) The cap brass is not provided with a steel keep

(2) The fork gap is relatively narrower owing to the piston rod nut having to deal with inertia only

(3) The brasses are of cast steel instead of gun metal

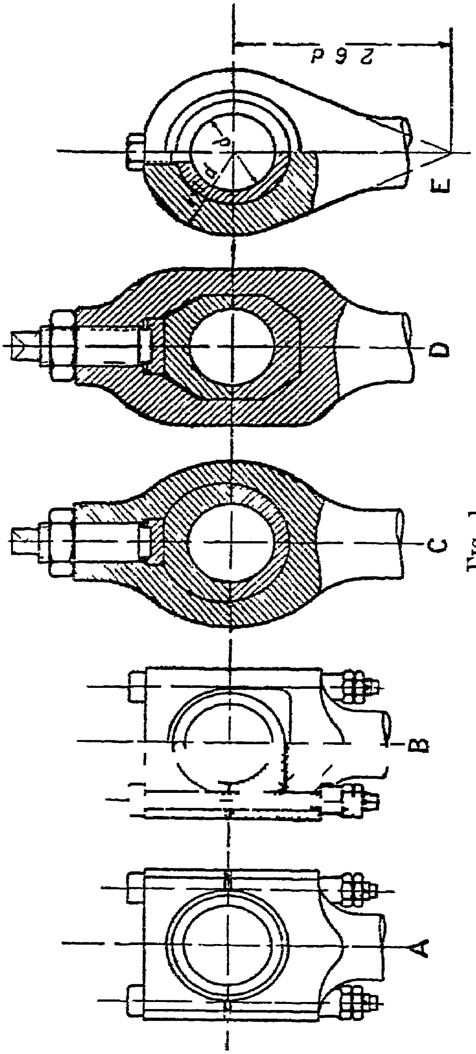

FIG 1

Points of Detail —The lubrication of the big and small ends has been referred to under crank shafts and pistons Where forced lubrication is used an oil hole is generally drilled in the rod to conduct the oil from the big to the small end in the case of high speed engines External pipes are used for this purpose in large slow running engines but are unsuitable for high speeds owing to the tendency of the joints to work loose

When air compressors oil pumps or other gear are worked by links from the connecting rod the connection to the latter should be made near the top end as in Fig 157 so that the strength of the rod to resist buckling is not impaired

Strength of Connecting Rods.—The forces acting on the connecting rod are:—

(a) The joint effect of

 (1) The pressure load on the piston.
 (2) The inertia of the piston and crosshead.
 (3) The piston ring friction.
 (4) The lubricated friction of piston and crosshead,

all divided by the cosine of the angle of obliquity of the rod.

(b) The longitudinal component of the inertia of the rod itself.

(c) The transverse component of the inertia of the rod itself.

(d) The friction of the top and bottom end bearings.

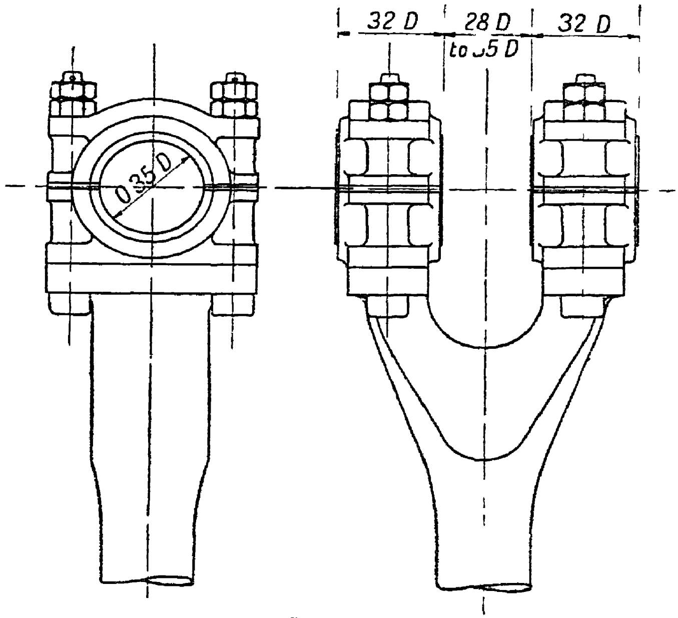

FIG. 156.

When considering the compressive stress of the rod on the expansion stroke one is on the safe side in neglecting item (4). The tensile forces attain their maximum at the top dead centre following the exhaust stroke, and the reciprocating parts being then at their position of minimum speed, item (4) may probably be neglected with safety.

Item (b) is estimated with sufficient accuracy by the usual

procedure of dividing the mass of the rod between the recipro cating and revolving parts in that ratio in which the centre of gravity of the rod divides its line of centres

Item (c) gives rise to a bending moment the maximum value of which is given approximately by the formula —

$$f = \frac{\left(\frac{n}{100}\right)^2 R\, L^2}{26d} \qquad (1)$$

Where f = Bending stress
n = Revolutions per minute
R = Crank radius in inches
L = Length of connecting rod in inches
d = Diameter of rod in inches (mean)

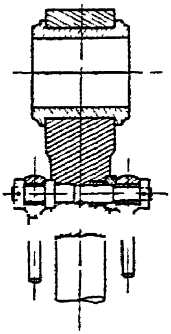

FIG 15

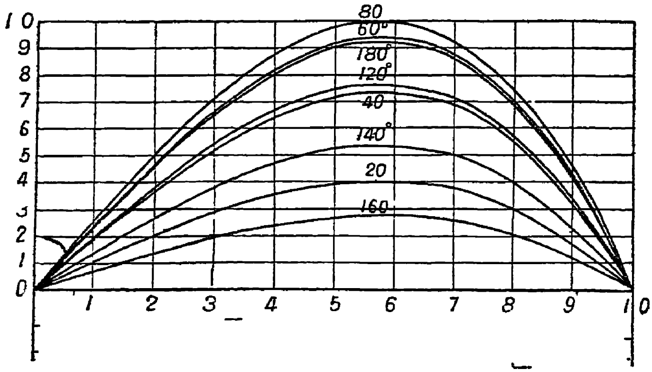

FIG 158

The sign of the bending moment is such that the latter always tends to bend the rod outwards to the side on which the crank stands The variation in the magnitude of the stress over the length of the rod for various positions of the crank relative to the top dead centre is shewn in Fig 158 for a rod five cranks long The stress varies as sin $(\theta+\phi)$ where θ = the crank angle relative to top dead centre and ϕ = the angle of obliquity of the rod and as $Lx - \frac{x^3}{L}$ where L is the length of the rod and x is the distance from the small end of the section under consideration

The assumptions made use of in equation (1) are that the rod is of uniform section and as is usually assumed in books

on applied mechanics and machine design that the influence of the rod ends is small

Item (d) may be estimated on the assumption that the co efficient of friction attains Morin s value of 0 15 for slightly greasy metal at the top or bottom end or at both ends simul taneously The effect of journal friction is to divert the line of thrust from the centre line of the rod and the amount of this deviation is found by the well known graphical construc tion shewn in Fig 159 in which the line of thrust is shewn

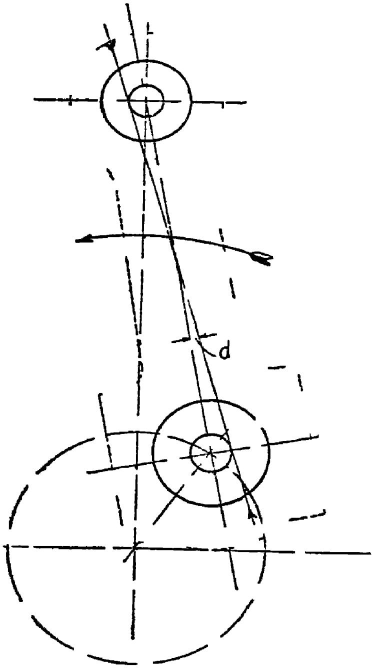

Fig 159

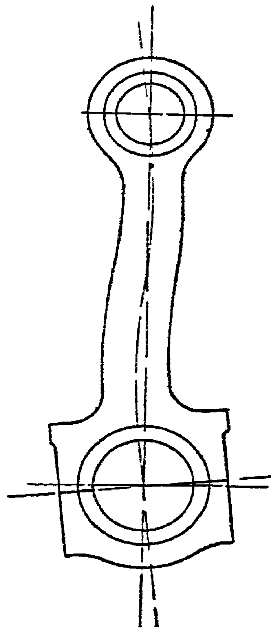

Fig 160

to be tangential to two very small circles whose radii are equal respectively to the radii of the crank pin and gudgeon pin multiplied by the coefficient of friction The deviation at any point of the rod of the line of thrust from the line of centres will be denoted d

The effect of this deviation is to bend the rod into an S shape as shewn much exaggerated in Fig 160 This form of failure is one consistent with the Eulerian theory of pure buckling but usually regarded as an improbable solution

The author has actually seen one instance of a Diesel Engine rod failing in this way and the effect was probably due to the causes indicated above arising in acute form To be on the safe side it seems advisable to consider two cases —

(1) Coefficient of friction negligible at the gudgeon pin and equal to 0 15 at the crank pin

(2) Coefficient of friction negligible at the crank pin and equal to 0 15 at the gudgeon pin

The weak sections are clearly those (cf Fig 163 KK and SS) where the big and small ends merge into the shaft with a

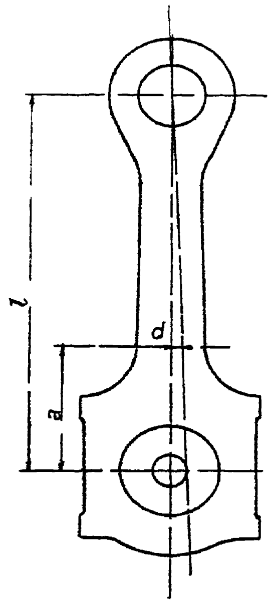

Fig 161

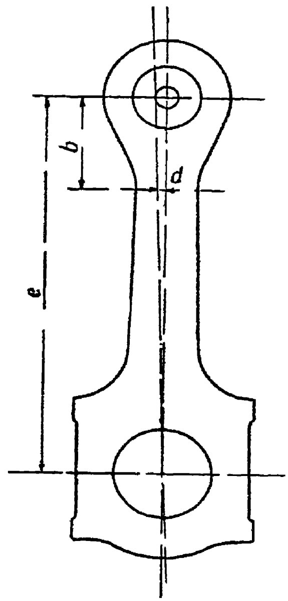

Fig 162

radius and incidentally those selected by the draftsman when giving dimensions for the diameters of the rod The reasonableness of making the rod tapered will now be apparent as the bending effect is clearly greater at the large end of the rod on account of the crank pin being of greater diameter than the gudgeon pin In all probability these considerations had nothing to do with the practice but illustrate a fact that has a very important bearing on the part of machine design viz that a construction which appears wrong to the eye of an individual gifted with a sense of form will usually on investiga

tion be found unsound in principle and vice versa This theme is one that might profitably be made the subject matter of a less specialised book than this but it may be worth mentioning here that the conscious or unconscious recognition of this principle plays a large part in mechanical design and the most thorough application of scientific principles seems always to result in graceful proportions

Returning to Fig 160 it is evident that the deflection at any point of the rod should to be strictly accurate be added to the deviation of the line of thrust at that point in order to find the bending moment and further this new bending moment involves the construction of a revised deflection curve and so on This evidently calls for some form of mathematical treatment which with certain approximations can readily be applied It will be found however that the deflections involved are small compared with the deviation of the line of thrust and whatever error may be incurred can be considered to be covered by the factor of safety

On these assumptions d for the weak sections KK and SS is given by the following —

Case (1) Fig 161 $$d = 0{\cdot}15\,P\,\frac{l-a}{l} \qquad (2)$$

Case (2) Fig 162 $$d = 0{\cdot}15\,R_g\,\frac{l-b}{l} \qquad (3)$$

And $$f = P\left(\frac{1}{A} + \frac{d}{Z}\right) \qquad (4)$$

Where P = The thrust in the rod in lb
A = Sectional area of rod in sq in
Z = Sectional modulus of rod in in ^{3}
f = Maximum compression stress in lb per sq in
R = Radius of crank pin
R_g = gudgeon pin

It will be noticed that the strut formulæ of Euler Gordon and Rankine and others have not been utilised above It appears to the writer that these formulæ are irrelevant to the case of Diesel Engine connecting rods for the following reasons —

(1) Euler s formula is based on the calculation of the load required to produce elastic instability and with short rods the stress commonly works out at a higher value than the ultimate strength

(2) The Gordon and Rankine formulæ are based on experimental values of the buckling stress under static conditions and give no indication of the strength under repetitions of stresses which are generally only a fraction of the buckling load

It seems more rational therefore to calculate the maximum direct stresses as closely as possible and to apply to the approximately known fatigue stress of steel a factor of safety of 2 5 to 3 which is known to be satisfactory in other cases

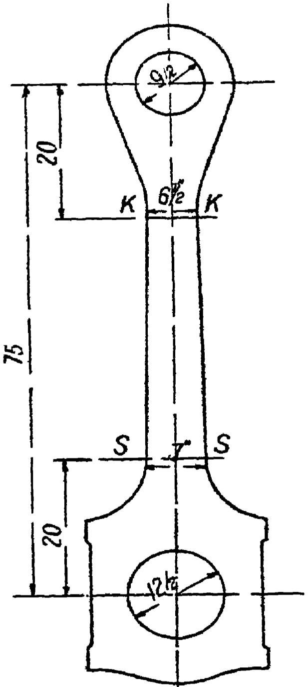

Fig 163

In view of occasional abnormal pressures of about 1000 lb per sq in it is interesting to see what factor of safety a given rod has for meeting such contingencies and the table of buckling stresses given on page 194 may be used for this purpose

Example of Stress Calculation for Connecting Rod —

FOUR STROKE CYCLE

Bore of cylinder	24 in
Stroke	30 in
Revolutions per minute	200
Weight of piston (trunk)	2200 lb
connecting rod complete	2500 lb

Main dimensions of rod as in Fig 163 under —

(1) Calculation of stress due to thrust 30 after firing centre

Piston pressure load $=0\ 785 \times 24^2 \times 500 = 226\ 000$ lb

Inertia load 30° after dead centre

$$=\left(\frac{2\pi\times200}{60}\right)^2\times15\times\frac{2200+0\ 35+2500}{386}\times(\cos 30\ +\tfrac{1}{5}\cos 60\ )$$

$$=51\ 000 \text{ lb}$$

Resultant vertical force $=226\ 000-51\ 000=175\ 000$ lb

At 30 after dead centre the obliquity of the rod is 6

Connecting rod thrust $=175\ 000 - \cos 6\ = 176\ 000$ lb $=P$

At section KK — Area $=33\ 2$ in $^2=A$

Section modulus $=27\ 0$ in $^3=Z$

Deviation of line of thrust $=\dfrac{0\ 15 \times 4\ 75 \times 55}{75}=0\ 52=d$

and the stress $f=176\ 000\left(\dfrac{1}{33\ 2}+\dfrac{0\ 52}{27\ 0}\right)=8970$ lb /sq in

At section SS —

$$\text{Area}=38\ 5 \text{ in}^2=A$$
$$\text{Section modulus}=33\ 7 \text{ in}^3=Z$$

Deviation of line of thrust $=\dfrac{0\ 15 \times 6\ 25 \times 55}{75}=0\ 69$ in

and the stress $f=176\ 000\left(\dfrac{1}{38\ 5}+\dfrac{0\ 69}{33\ 7}\right)=176\ 000 \times 0\ 0464$

$$=8160 \text{ lb /in}^2$$

(2) Calculation of stress due to inertia bending at 30 after dead centre

Maximum inertia stress in rod from equation (1)

$$=\frac{2^2 \times 15 \times 75^2}{26 \times 6\ 75}=1920 \text{ lb /in}^2$$

From Fig 158 the fraction of this maximum applying to position KK at 30 after dead centre is 0 37

Inertia bending stress at section KK $=0\ 37 \times 1920$

$$=710 \text{ lb /in}$$

The fraction applying to section SS at 30 after dead centre is 0 49

Inertia bending stress at section SS $=0\ 49 \times 1920$

$$=940 \text{ lb /in}^2$$

(3) Resultant stress at KK $=8970-710=8260$ lb /in ^{2}

Since the bending actions due to inertia and eccentricity of thrust are of opposite sign

Resultant stress at SS $=8160+940=9100$ lb /in ^{2}

Since the two bending actions are of the same sign

(4) Tensile stress at SS at beginning of suction stroke

$$\text{Inertia force}=\left(\frac{2\pi \times 200}{60}\right)^2 \times 15 \times \frac{2000+0\ 35 \times 2500}{386} \times 1\ 2$$

$$=63\ 200 \text{ lb}$$

Stress at SS $=63\ 200\left(\dfrac{1}{38\ 5}+\dfrac{0\ 69}{33\ 8}\right)=2930$ lb /in ^{2}

The total range of stress is therefore

$$9100+2930=12\ 030 \text{ lb /in}^2$$

The range of stress required to produce fracture of mild steel by fatigue appears to be about 35 000 lb /sq in so the factor of safety is about 3

Calculation on the above lines might with advantage be made for several different positions of the crank

It is evident that the results of the calculation depend very largely on the assumed conditions of journal friction but it should be borne in mind that almost any possible combination of unfavourable conditions is a probable contingency in the combined lives of a number of similar engines

Proportions Found in Practice —In the preceding example the mean diameter of the rod is approximately 0 28 of the diameter of the cylinder a very favourite ratio in practice In different designs this ratio varies from about 0 26 to 0 33

The maximum and minimum diameters are usually about 5 % more and less than the mean

Connecting Rod Bolts —In four stroke engines these are usually proportioned to the maximum inertia load with a nominal stress of 4000 to 6000 lb /in ^{2} based on the inertia and centrifugal loads divided by the area of two bolts at the bottom of the threads With trunk piston engines failure when it occurs is generally due to piston seizure to which it would be difficult to apply definite rules of calculation Danger of seizure is largely eliminated by the use of a cross head The strength of connecting rod bolts for four stroke Diesel Engines forms the subject matter of a paper by Mr P H Smith read by him before the Diesel Users Association and containing the results of several years experience For the big end it appears that the bolts seldom fail if made of a diameter 12 to 13 % of the cylinder bore For the small end if bolts are used at all the only safe rule is to make the bolts as large as the space available will allow Mr Smith also points out that the bolts for both big and small ends are not equally stressed as may easily be seen by reference to Fig 164

Owing to the deviation of the line of pull from the centre line of the rod that bolt (No 1 in the Fig) which first passes the top dead centre at the beginning of the suction stroke is nearer the line of pull than the other bolt and consequently more highly stressed

If P =Pesultant pull in lb
S =Centres of bolts in in
d =0 15 radius of crank pin

Then

$$\text{Pull in bolt No. } 1 = \frac{P \times \left(\frac{S}{2} - d\right)}{S}$$

$$\text{Pull in bolt No. } 2 = \frac{P \times \left(\frac{S}{2} - d\right)}{S}$$

With crosshead engines the small end bolts have to carry the inertia load due to piston, piston rod and crosshead, and also any frictional forces acting on the piston and crosshead. The

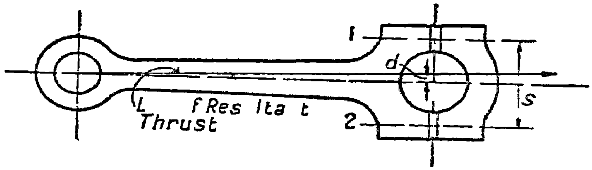

FIG. 164

latter being more or less indeterminate, it is customary to allow a nominal stress on these bolts about 30% less than that allowed for the big end bolts. The bolts or studs connecting the crosshead to the piston rod and the latter to the piston are given a large margin of strength for the same reason. Connecting rods for two stroke engines are not as a rule distinguishable from those for four stroke engines, as the possibility of compressions being lost has always to be kept in view.

Indicating Gear.—The only satisfactory gear for obtaining accurate cards consists of a link motion directly connected to the piston. The usual arrangement is shewn diagrammatically in Fig. 165 for both trunk and crosshead engines. The conditions for giving an accurate reproduction of the motion of the piston are:—

(1) The line of the cord to be at right angles to the mean position of the short arm of the lever.

(2) The long arm of the lever to make equal angles of swing above and below the horizontal.

(3) The versine of the arc of swing of the drag link to be negligible in comparison with the stroke of the engine.

The mechanical details of indicator gears are hardly of sufficient interest to require description here, but it may be

well to mention that indicating is of far more importance in the successful running of a Diesel Engine than in that of a steam or even a gas engine and consequently all the more care should be given to the design of the gear by which the indicating is accomplished Makeshift or temporary gear should not be tolerated but the same attention paid to lubrication and bushing of joints etc as on other parts of the engine

Literature — Piston Cooling for Diesel Engines Article in *The Motor Ship and Motor Boat* July 18th 1918 *et seq*

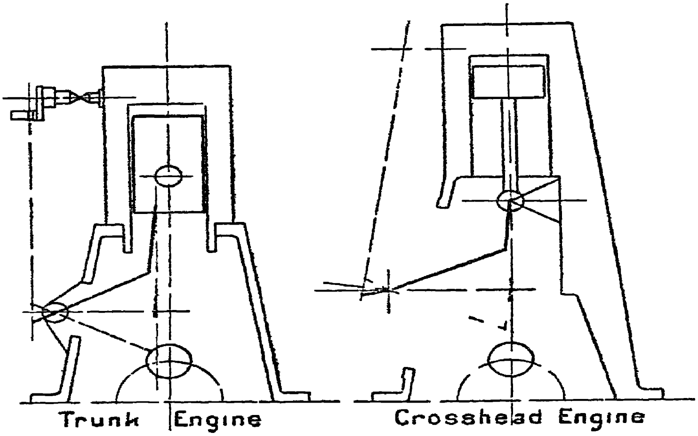

FIG 165

CHAPTEP XI

FUEL OIL SYSTEM

For puiposes of description the complete fuel oil system is conveniently divided into two parts the first consisting of those elements such as tanks etc which are external to the engine and the second of those organs of the engine itself which are directly concerned with the delivery of the fuel to the working cylinder

External Fuel Oil System —Fig 166 repre ei ts diagiam matically a fuel system for a small Diesel power station and

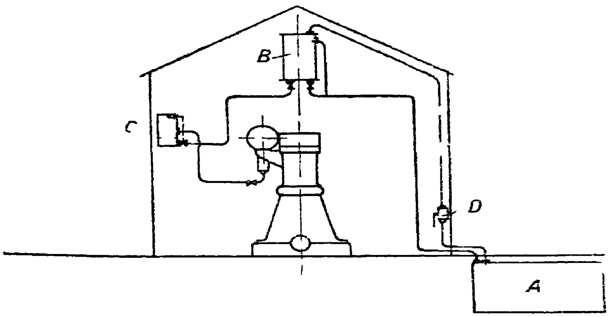

Fig 166

consists essentially of a main stoiage tank A a ready use tank B a filter C and a pump D for raising the oil from the storage tank

The stoiage tank is preferably arranged undeiground as close as possible to the railway siding so that oil can be run from the railway tank waggon to the storage tank by gravity through a hose pipe Some foim of level indicator or a plugged hole for a sounding rod should be provided The capacity of the tank will depend on the size of the station and the local conditions of supply

The pump D by means of which the oil is pumped to the ready use tank may be of the semi rotary type capable of being worked by one man in the case of small stations but where the daily demand is greater a motor driven rotary or reciprocating pump is generally fitted

The ready use tank may have a capacity of say half a day s run so that the routine of replenishing it will occur twice daily Some form of float indicator should be fitted so that the level of oil may be conveniently ascertained from the engine room floor Other necessary fittings are an overflow pipe leading to the storage or a special drain tank and a drain valve communi cating with the overflow pipe The tank must be closed at the top to exclude dirt The valves in connection with the fuel system are preferably of the gate or sluice type as cocks are liable to leakage and globe valves tend to choke by accumula tion of sediment The tank only requires to be located a few feet above the level of the filter as the discharge is very small

The filter usually consists of a cylindrical tank of about 40 gallons capacity located about two feet above the level of the cylinder cover and provided with a filter diaphragm at about a third of the height of the tank from the bottom The diaphragm consists of a sheet of felt sandwiched between two sheets of wire gauze and reinforced by an angle iron ring

The fuel enters the filter at the bottom passes through the diaphragm by virtue of its static head and is drawn off by the engine fuel pump at a point a few inches above the diaphragm The filter vessel is prevented from being overflooded by a ball float mechanism which closes the inlet cock when the oil reaches a predetermined level The plug of this cock is kept fairly tight by means of a spring acting on the plug but slight leakage is almost inevitable so it is desirable to mount the filter on an oil tight tray provided with a drain It is very usual to provide a small reservoir of the same capacity as the filter arranged alongside the latter for the reception of paraffin by means of which the piping leading to the engine and also the fuel pumps and fuel valves etc may be cleansed from time to time by running the engine for a few minutes on this fuel before stopping the engine

Marine installations follow on similar lines with a few com plications The double bottom is used as a storage tank and the fuel is raised to the ready use tank by motor driven pumps when electric power from auxiliary engines is continuously

available or by means of pumps driven off the main engine in cases where the main engine drives its own auxiliaries In either case it is usual to install duplicate pumps to guard against breakdown The motion of ships being unfavourable to the successful operation of float devices the level of the oil in the ready use tank has to be inferred from gauge glasses test cocks and the like For similar reasons the filters must be totally enclosed and provided in duplicate with change over cocks so that they may be overhauled at any time In addition special requirements of the Board of Trade and Lloyd s have usually to be complied with

Fuel System on the Engine itself —The commoner arrangements fall into one of two broad classes —

(1) Those in which each cylinder has a separate fuel pump or separate plunger and set of valves to itself In this case the oil is delivered direct from the pump to the injection valve by the most direct route possible

(2) Those in which one fuel pump plunger upplies the oil for a plurality of cylinders usually a maximum of four In this case the pump delivers to a fuel main provided with a branch and distributing valve separate to each cylinder whereby the amount of oil delivered to each cylinder may be equalised while the engine is running

Engines of six or eight cylinders are divided into two groups of three or four so far as the oil system is concerned With marine engines the first system is at present the favourite and has the advantage that the failure of one pump does not affect the working of the remaining cylinders With the second system a stand by pump is provided ready to take over duty at a moment s notice

For land engines the two systems appear to be on an equality as nearly as can be ascertained by the reputations of representatives of both classes but if anything system (1) appears to be slightly the more popular of the two

Figs 167 and 168 illustrate the two systems diagrammatically It will be noticed that in Fig 167 the governor operates on all the pumps by means of a shaft extending nearly the whole length of the engine and as the quality of the governing is dependent on the freedom from friction of the governing mechanism it is desirable to mount this shaft on ball bearings The expense of providing separate pump bodies

and drives is sometimes reduced by grouping the pumps in the neighbourhood of the governor even to the extent of driving all the plungers by a common eccentric

With the arrangement of Fig 168 there is only one pump to regulate and this renders possible the use of a type of governor

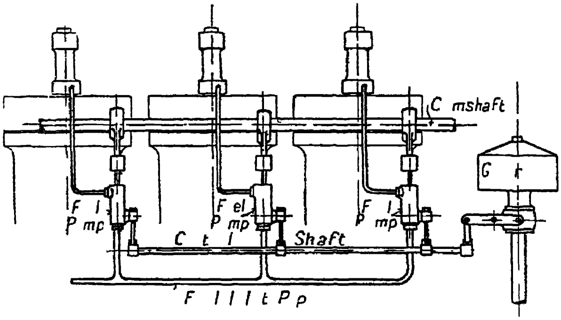

FIG 16

which is probably unrivalled for sensibility and which will be described later The distributors indicated in Fig 168 are a special feature of this system and are illustrated to a larger scale in Fig 169 A particularly neat arrangement of piping is obtained by combining the fuel distributor and blast air T piece in one fitting

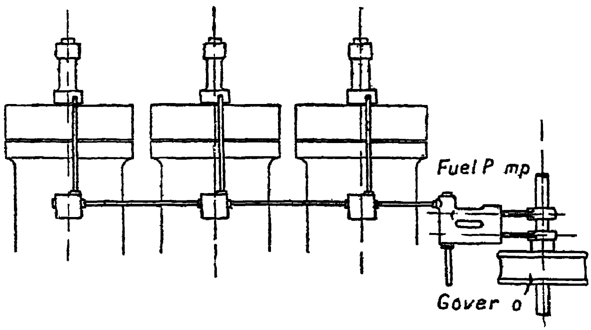

FIG 168

The inclusion of a non return valve prevents in a great measure the oil being forced back through the pump by the blast air pressure in the interval elapsing between the turning on of the blast air and the attainment by the engine of full working speed A non return valve is sometimes fitted to the

fuel valve itself for the same reason Vent cocks are provided on the distributors and sometimes on the fuel valves to enable the pipes to be primed before starting the engine

The priming may be effected in various ways —

(1) By gravity means being provided for holding the fuel pump valves off their seats during the process

(2) By means of an auxiliary hand operated and spring returned plunger on the fuel pump

(3) By means of the fuel pump plunger itself where provision has been made for disconnecting the latter from its operating eccentric in order to enable it to be operated by a hand lever provided for the purpose

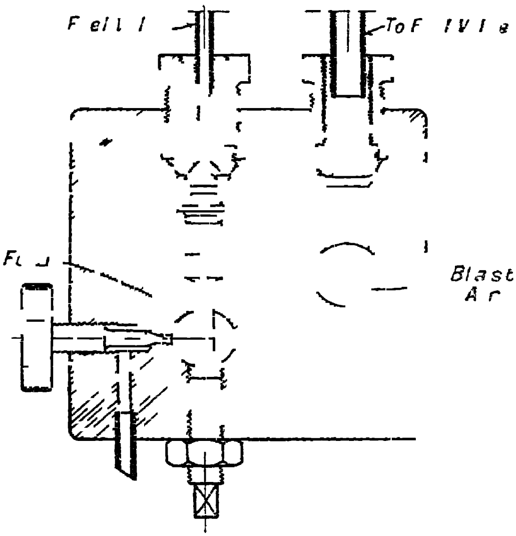

FIG 169

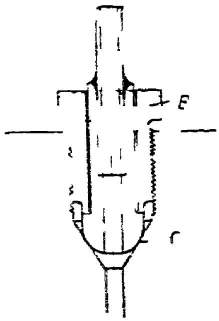

FIG 1 0

The piping in connection with the high pressure fuel system deserves special attention on account of the high pressures used and the type of union shewn in Fig 170 is probably the most satisfactory that has yet been devised both for oil and high pressure air

Fuel Pumps —A simple fuel pump for a large marine engine is shewn in Fig 171 and is representative of a large class of pumps for both marine and stationary purposes

The operation of the pump is almost obvious from the figure but may be described briefly as follows —

The eccentric A works the plunger B which is guided at C E and F are the delivery and suction valves respectively and the latter communicates with the suction chamber D to which the fuel is led by means of a pipe not shewn in the figure M is an auxiliary plunger operated from the crosshead C by links I H K etc and whose function is to keep the suction valve off its seat for a fraction of the delivery stroke depending upon how much oil is required per stroke The duration of this inoperative portion of the stroke is altered as required by raising or depressing the point K by means of an eccentric keyed to the shaft L according as less or more oil is required In the case of a governed engine the shaft L is controlled by the governor On marine engines the shaft L is operated by hand gear consisting of levers rods etc Neglecting the obliquity of the eccentric rod the main plunger describes simple harmonic motion of amplitude equal to half the stroke of the eccentric and it will be clear from the drawing that the auxiliary plunger M will describe a similar motion exactly in phase with the first but of amplitude equal to stroke of main plunger $\times \dfrac{KH}{KI}$

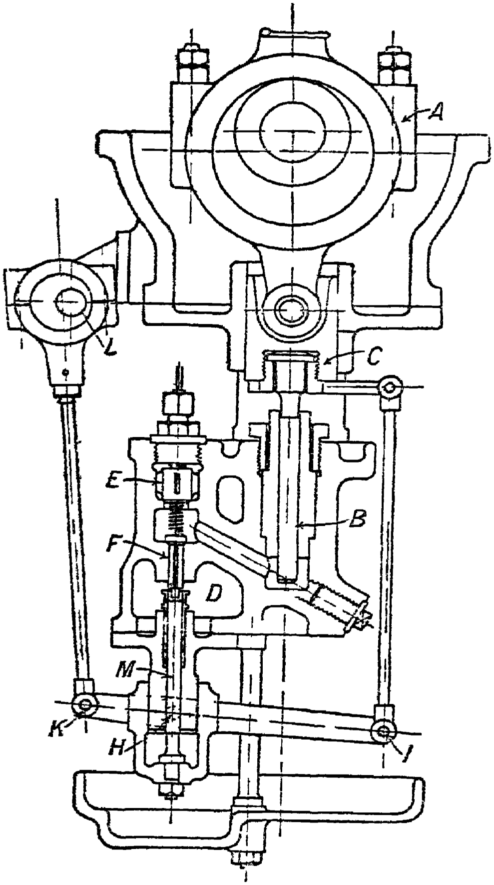

Fig 171

When the main plunger is at the bottom of its stroke the auxiliary plunger is also at the lowest point of its travel and the clearance between the top of the auxiliary plunger and the suction valve multiplied by the ratio $\dfrac{KI}{KH}$ is equal to the effec

tive stroke of the pump that is that portion of the stroke during which the suction valve is on its seat as of course it must be (apart from viscosity effects) for delivery to take place The quantity of oil delivered per stroke therefore depends on a certain clearance between the auxiliary plunger and the suction valve which clearance is readily adjusted by shortening or lengthening the rod LK when assembling or adjusting the engine and in the ordinary course of running by the eccentric at L

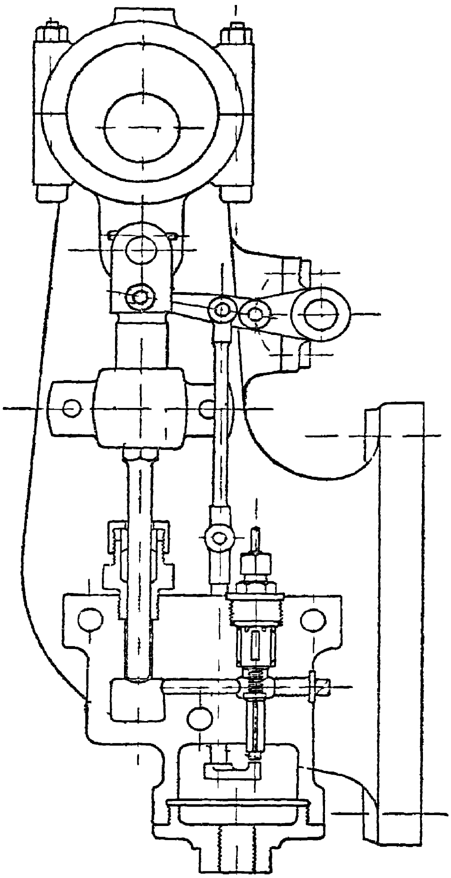

FIG 172

Constructional Details—The pump body plunger sleeve and guide are of cast iron The main and auxiliary plungers the crosshead pin and joints in the linkwork are of case hardened steel The valves may be either of steel or cast iron If the latter then the suction valve should be fitted with a hardened steel thimble where it makes contact with the auxiliary plunger The main eccentric and strap may be of cast iron the lower half of strap being white metalled in some cases It will be noticed that no packing is provided for the main plunger but reliance has been placed on the fit of the plunger With good workmanship the leakage should not be excessive[1] A cast tray is provided to catch drips during working and the overflow at priming A light sleeve encircling the auxiliary plunger is arranged for operation by external gear so that the suction valve may be lifted by an emergency governor in cases of excessive speed and also by hand in case it is desired to cut any individual cylinder out of operation

[1] Except with tar oil for which this arrangement is unsuitable

Variations of this system embodying the same principle are shewn in Figs 172 and 173 The front view of the latter shews three pumps grouped together but each worked by its own eccentric Fig 174 shews four plungers being operated by eccentrics in common It is evident that with this arrangement the oil delivered to the pulverisers of the various cylinders will have different allowances of time in which to settle before injection into the cylinder This appears to have no effect on the efficiency but it is usual to space the eccentrics so that oil is not in process of delivery whilst a fuel valve is open The pumps so far illustrated have been driven off the cam shaft That shewn in Fig 175 is arranged with horizontal plungers

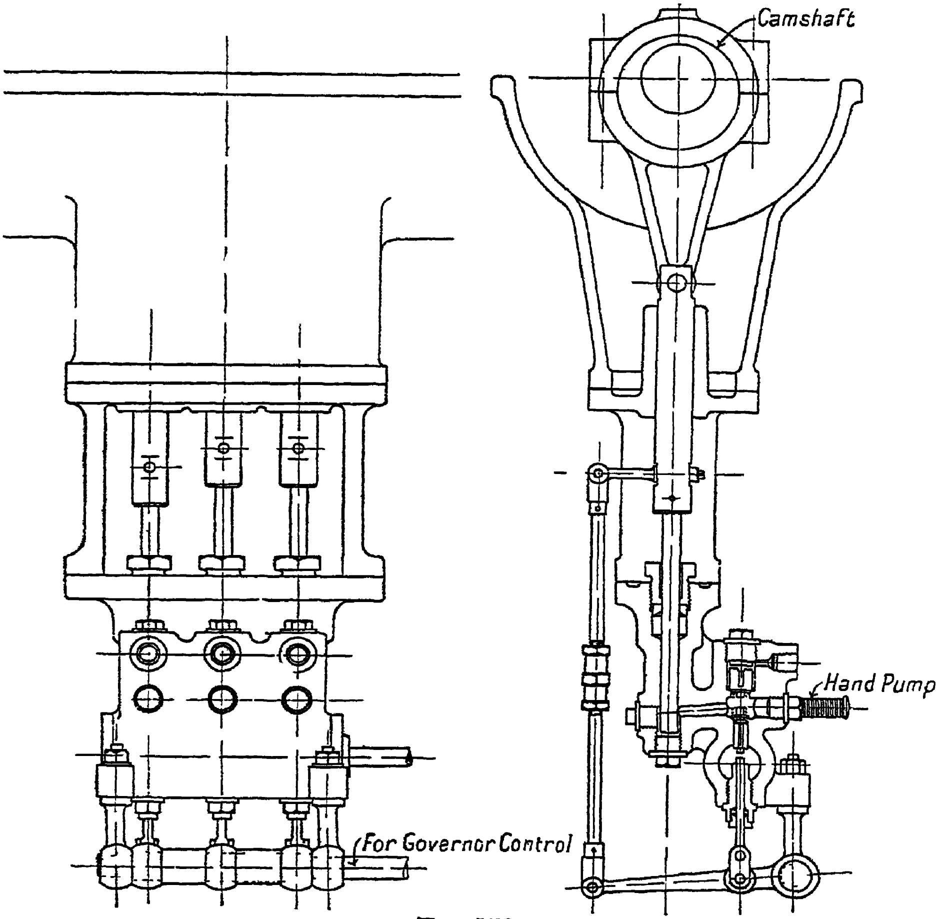

FIG 173

for driving off a vertical shaft The auxiliary plunger is driven by a separate eccentric which on account of the intermediate lever L requires to be at 180 degrees or thereabouts to the main eccentric The suction valve control may in this case be effected in one of two ways

(1) By an eccentric movement of the lever L
(2) By advancing or retarding the auxiliary eccentric

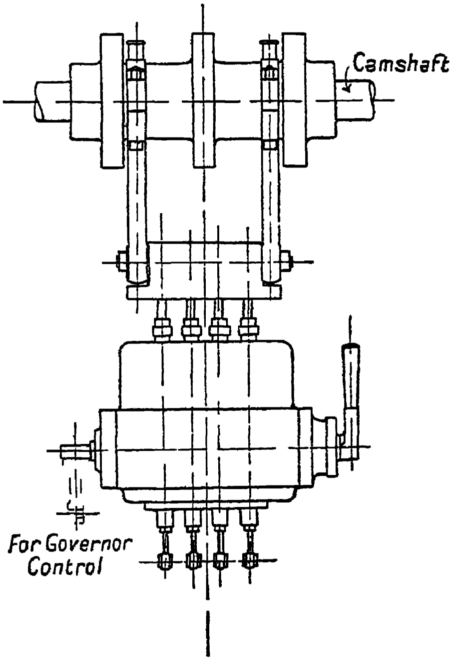

FIG 174

The latter leads to a veiv neat and efficient arrangement of governor and fuel pump to which reference has already been made It will be immediately obvious that with a given maximum clearance between the suction valve and the auxiliary plunger an angular movement of the auxiliary eccentric will have the effect of advancing or retarding the instant at which the suction valve comes on its seat and con sequently increasing or decreasing the amount of oil delivered per stroke This angular movement is effected very simply by a type of governor which has been well known for a long time in steam practice and which is illustrated in Fig 184

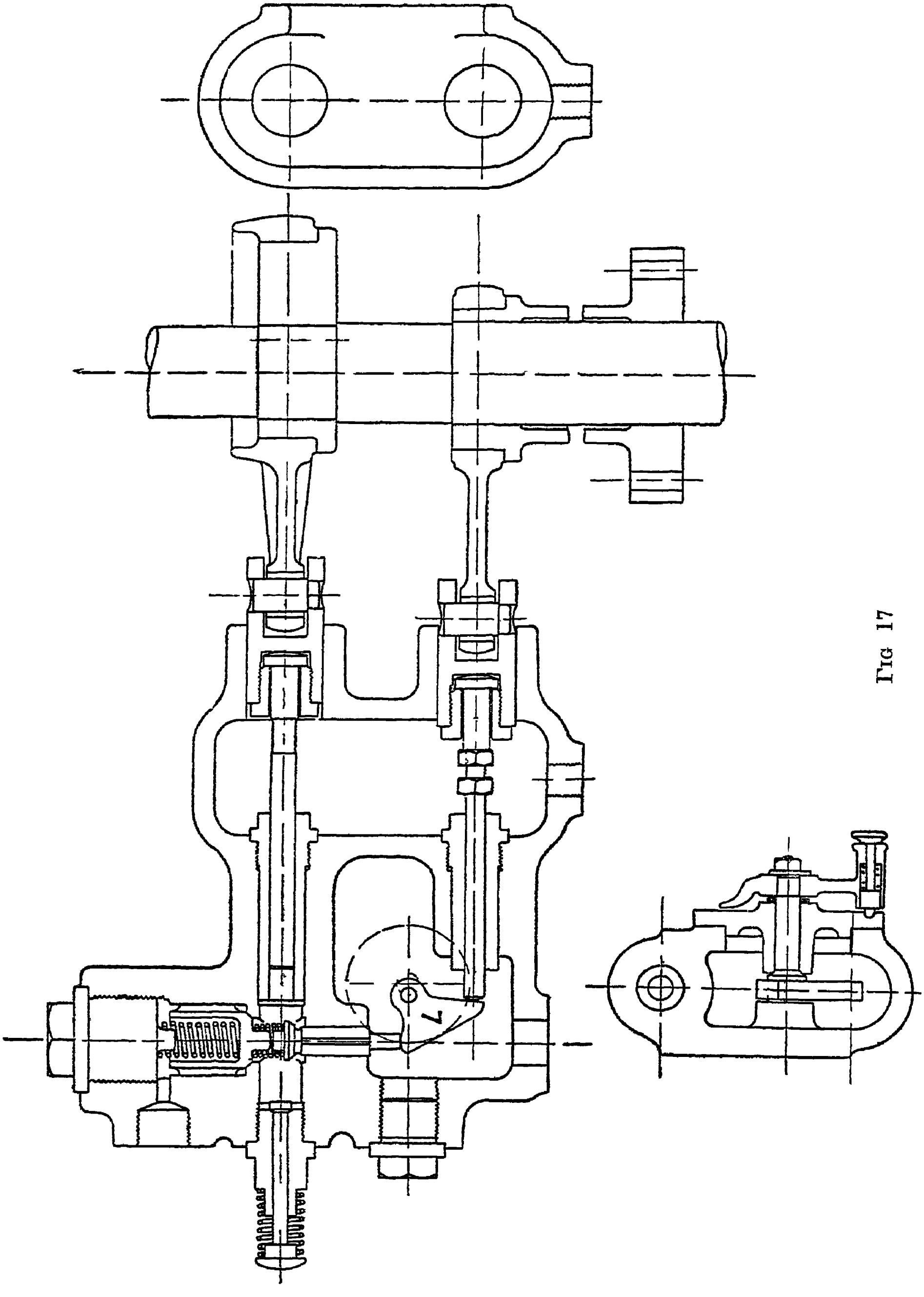

Fig 17

Returning to Fig 175 the use of this type of fuel pump is almost entirely confined to land engines The provision of an eccentric mounting for lever L enables the pump to be set in three different positions apart from its normal running position viz —

(1) Starting In this position the lever is moved so that the maximum clearance under the suction valve is increased about 50% so that the delivery of oil per stroke is increased correspondingly

(2) Stop In this position the suction valve is held continuously off its seat and no oil is delivered

(3) Priming In this position both suction and delivery valves are held off their seats and the oil has a clear passage through the pump

Figs 177 and 178 will make this matter clear without further explanation

Details of Fuel Pumps —The bodies are usually of cast iron but solid slabs of steel are sometimes used In designing the body three considerations should be kept in view —

(1) The shape to be favourable to sound casting

(2) As few machining operations as possible to be necessary apart from those which can be done on a drilling machine

(3) The pump chamber and passages to be free from air locks

Owing to the costly precautions necessary to ensure the plunger and guide being concentric and in line it is convenient to allow some side play at the point where they join as in Fig 179 Some different forms of plunger packing are shewn in Fig 180 and a selection of suction and delivery valves in Fig 181 Fig 182 shews a hand operated plunger for priming purposes

Calculations for Fuel Pumps —The process of computing the capacity of a fuel pump for a proposed engine is most easily illustrated by an example as follows —

B H P of one cylinder (four stroke) 250
One plunger to each cylinder
Estimated fuel consumption 0 4 lb per B H P hour
Revolutions per minute 120

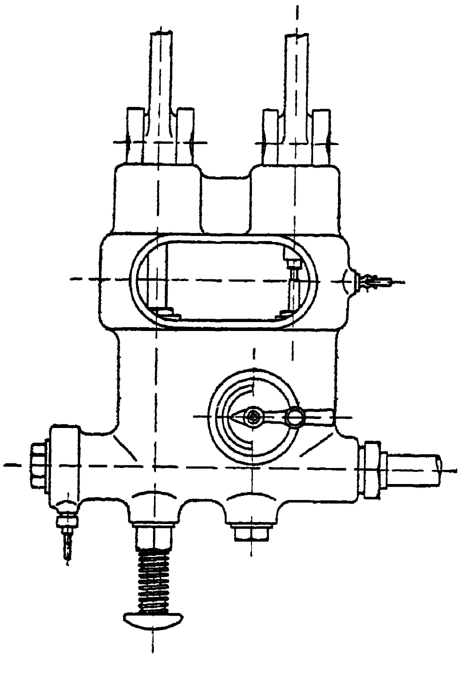

FIG 176

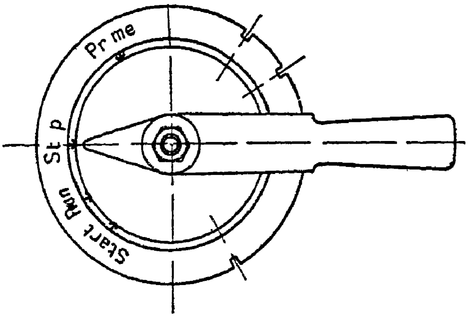

FIG 177

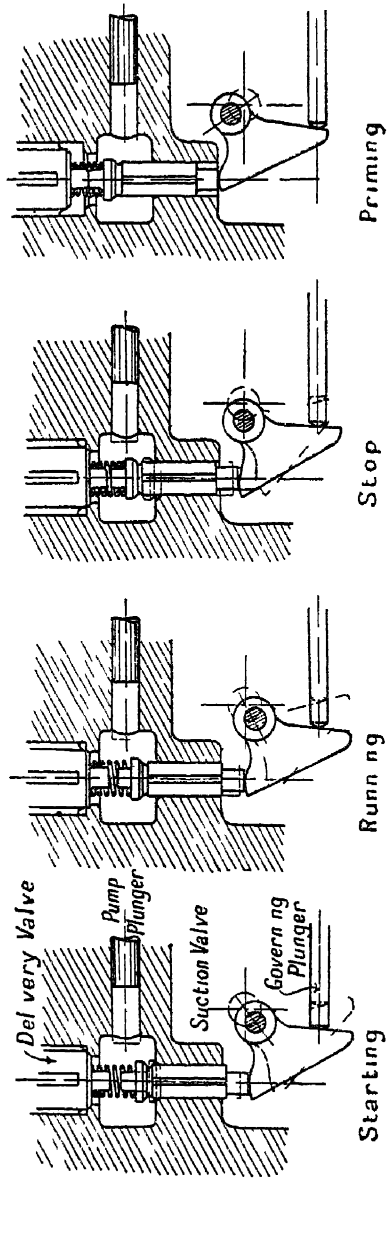

FIG 178

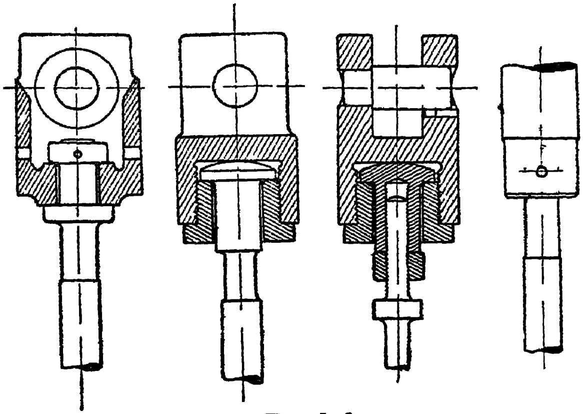

FIG 179

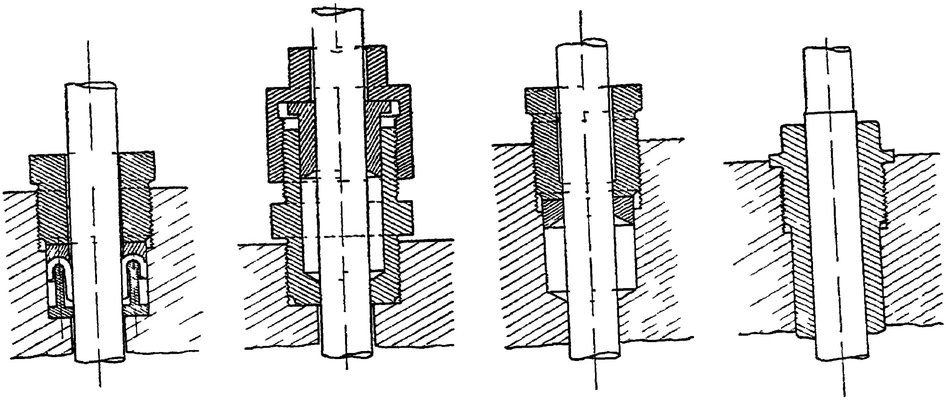

FIG 180

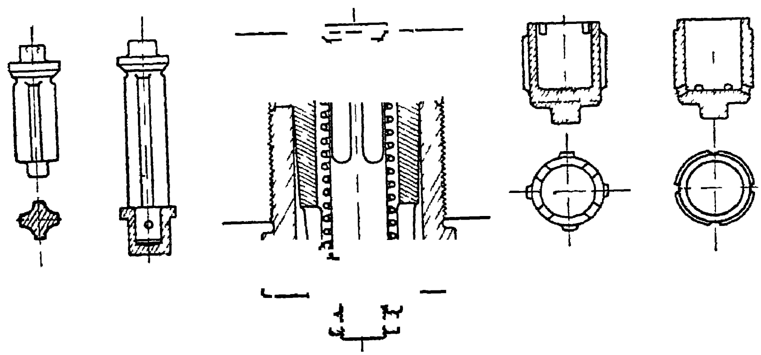

FIG 181

Estimated quantity of fuel per cycle $= \frac{0\ 4 \times 250}{60 \times 60} = 0\ 0278$ lb

Volume occupied by 1 lb of fuel = about 31 cub in

Therefore volume of fuel per cycle $= 0\ 0278 \times 31 = 0\ 86$ in ^{3}

Adding 50 % to allow for overload possible increase of fuel consumption leakage of plunger etc —

Stroke volume of plunger $= 1\ 5 \times 0\ 86 = 1\ 29$ in ^{3}

Which is satisfied by a plunger diameter of $\frac{3}{4}$ in $\times 3$ in stroke

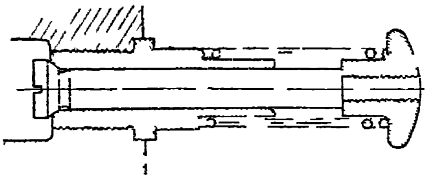

FIG 182

This size of plunger would only be permissible on a marine engine If the cylinder belonged to a governed engine the stroke volume of the fuel pump plunger would need to be about four times the above figure as it is found that good governing at all loads is only to be obtained by using about the last quarter of the stroke This is probably due to the fact that the quantity of fuel consumed by the engine in a given time is not proportional to the load but more nearly propor tional to the load plus a constant representing the engine friction The actual position taken up by the governor and the effective stroke of the pump plunger at any specified pro portion of full load are not easy to determine experimentally with great accuracy but the angular positions indicated in Fig 183 with reference to the fuel pump eccentric circle are those generally used as the basis of calculation

When one plunger is used to supply several cylinders the length of effective discharge period is limited by the condition that the latter should not overlap the injection periods In estimating the capacity of a fuel pump driven off a vertical shaft the speed of the latter must be kept in mind being usually the same speed as the engine and in some cases 50 % more

The valves hand plungers etc are suitable subjects for distributive standardisation For example a suction valve $\frac{5}{8}$ in in diameter would be quite suitable for all sizes of cylinder (assuming one plunger per cylinder) up to about 20 in bore provided that the use of fuels of exceptional viscosity were not

contemplated For oils like crude Mexican of the consistency (when cold) of tar larger valves are probably advisable With the valve arrangements in common use the diameter of the delivery valve is determined by that of the head of the suction valve plus adequate clearance for the flow of the oil round the latter

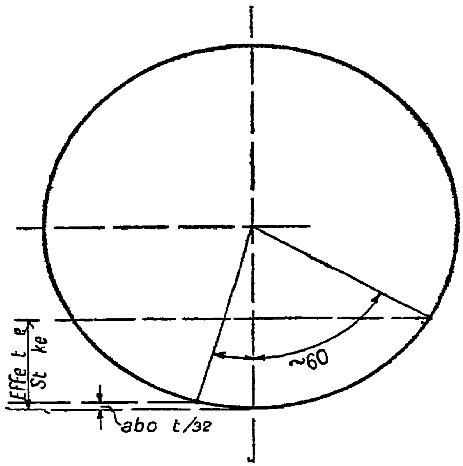

Fig 183

The general thickness of metal of the pump body is usually kept as uniform as possible to facilitate casting and the nominal stress in the neighbourhood of the pump chamber based on a blast pressure of 1000 lb per sq in is about 2500 lb per sq in The design of a fuel pump affords ample scope for a draftsman s skill in many directions in which numerical calculation plays a very small part and the following suggestions are offered —

(1) The arrangement generally to be neat and substantial and presenting an external appearance in keeping with its surroundings

(2) The flanges and brackets by which it is secured to the framework of the engine to be unobtrusive and to have the appearance of growing as naturally as possible out of the main body of the casing so as to convey an impression of rigidity and equilibrium

(3) Every detail to be carefully studied both with regard to its special function and also to economy in manufacture efficiency always having precedence over economy In particular case hardening and bushing of parts subject to wear must not be stinted and provision should be made for lubrication of all moving parts

(4) Valves and other internal mechanism to be easily accessible for inspection and overhaul

(5) Arrangements to be made to catch all drips both of fuel and lubricating oil avoiding the use of trumpery sheet iron guards and the like

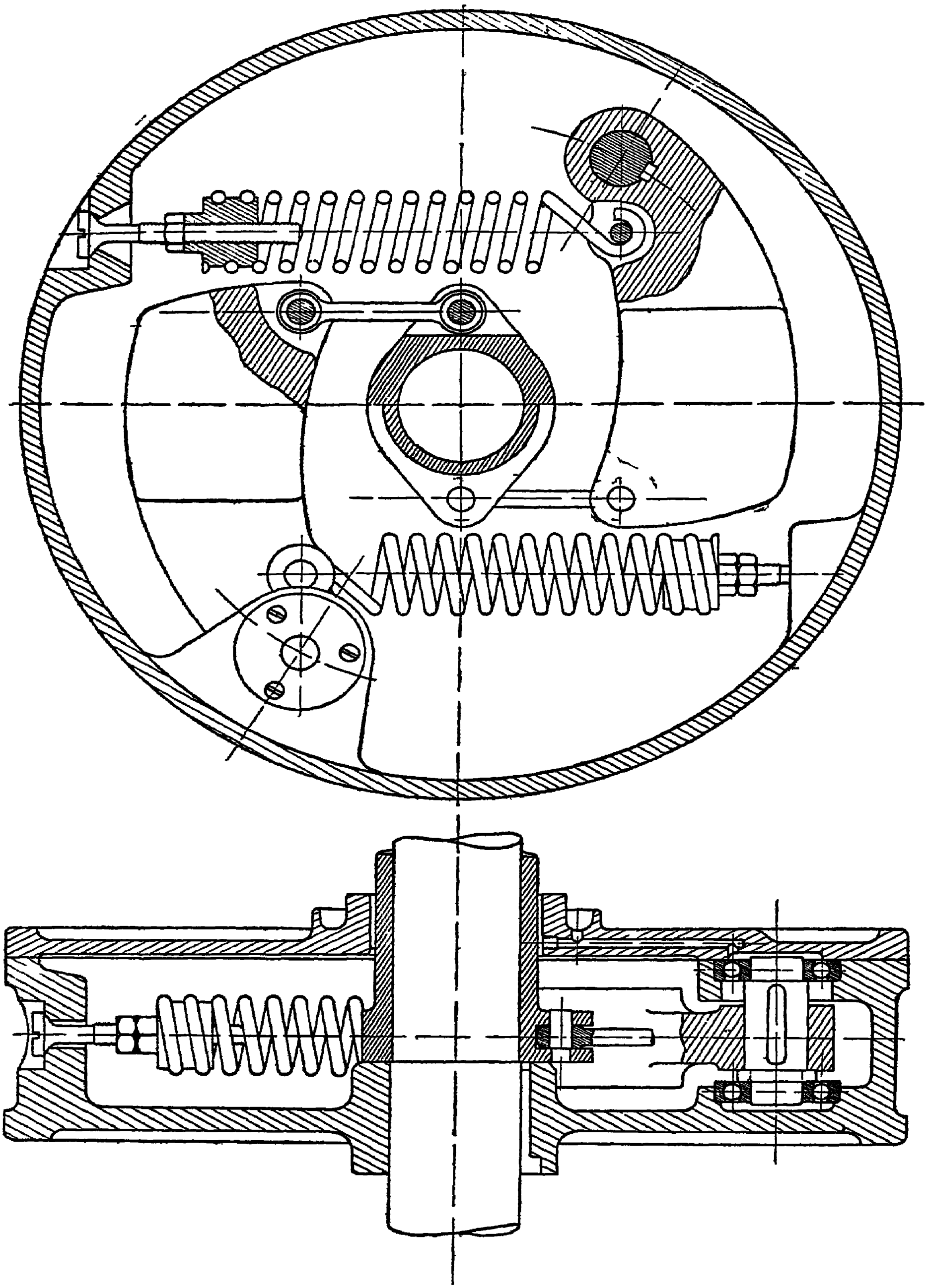

FIG 184

Many of the above principles apply of course to the design of any part of any high grade machine and they are mentioned here because the matter on hand provides an excellent oppor tunity of emphasising them in a particular case in which the subject is singularly free from the complications arising from calculations When the discussion is transferred to some large part of a machine in which the stresses are approximately determinate and the scope of the design appears to be limited by adjacent parts it becomes increasingly difficult to reconcile the ideals of high class design with the requirements of efficiency and economy and the skill of a designer may be gauged by the extent to which this difficulty is overcome From this point of view no part of a design can be said to be finally determined until the whole design is complete as there is always the possibility that a design for a certain part perfect in itself may require to be modified subsequently on account of its relationship perhaps remote to some other part as yet undetermined

Governors —It is not proposed to deal here with governor design generally as that is a subject for a specialist in this particular department of mechanical design but only to illustrate the application of governors to stationary Diesel Engines by means of a few examples and to give the main lines of calculation for the type of governor shewn in Fig 184 which is a type not usually standardised by governor specialists The action of the weights in causing rotation of the central sleeve will be immediately obvious from the figure The amount of this rotation between the limits of no load and full load should correspond with the angle $\sim$ 60 of Fig 183 but as a safety pre caution it is advisable to give the governor sufficient range to give a complete cut out and the sleeve should therefore be free to describe an angle of about 70 The first stage in the design of the governor is to rough out a drawing similar to Fig 184 fulfilling all the requirements as to space accessi bility etc and in which the above angular rotation is secured As regards the size of the governor it is generally wise to avail oneself of all the space obtainable The next step is to find the mass and centre of gravity of the weights and the positions of the latter in the extreme in and out positions A diagram similar to Fig 185 should now be constructed in which the abscissæ are distances of the weight from the centre of the shaft in inches and the ordinates are the centrifugal forces at

these distances at no load speed and full load speed respectively Point P corresponds to no load speed and no load distance from centre and point Q the same quantities for full load The line PQ then determines the properties which the controlling spring would have to possess if it were connected to the weight at its centre of gravity These properties are as follows —

(1) The initial tension when the weights are full in is equal to the centrifugal force corresponding to the point Q

(2) The weight of the spring per inch extension is equal to the slope of the line PQ that is the amount in lb by which the ordinate increases as the abscissa increases by one inch

Actually the spring is attached to the weight at a point nearer to the fulcrum than the centre of gravity and both the initial tension and the rate as found thus require to be increased in the ratio $\frac{k}{l}$ where —

k = Distance of the weight fulcrum from the line joining the C G of the weight to the centre of the governor

l = Distance of the weight fulcrum from the line of action of the spring

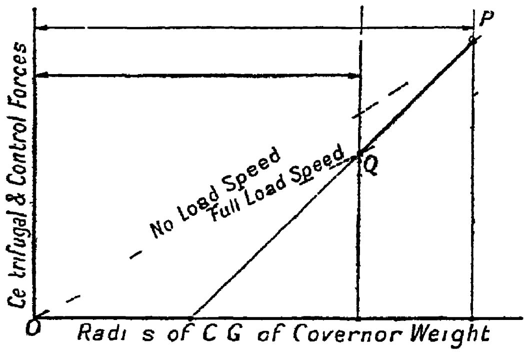

FIG 185

This very simple construction repeated as often as may be necessary in the process of trial and error contains all the dynamical calculation required to ensure sensibility and stability but it is advisable to provide adjustments for spring tension in the manner shewn in the figure to allow for un

avoidable errors and routine adjustment on the test bed Strictly speaking the diagram shewn in Fig 185 should be corrected to allow for the versed sine of the arcs described by the points of suspension and so on but these are practically negligible Other types of governor are designed on similar lines but are usually complicated by link mechanism of which the variations of configuration are not negligible

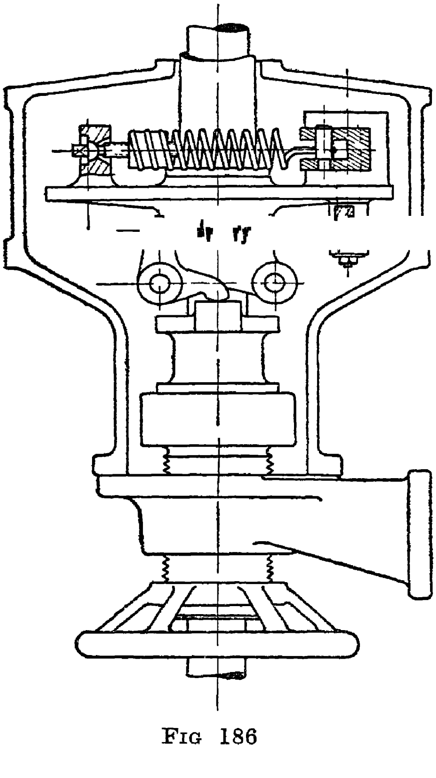

FIG 186

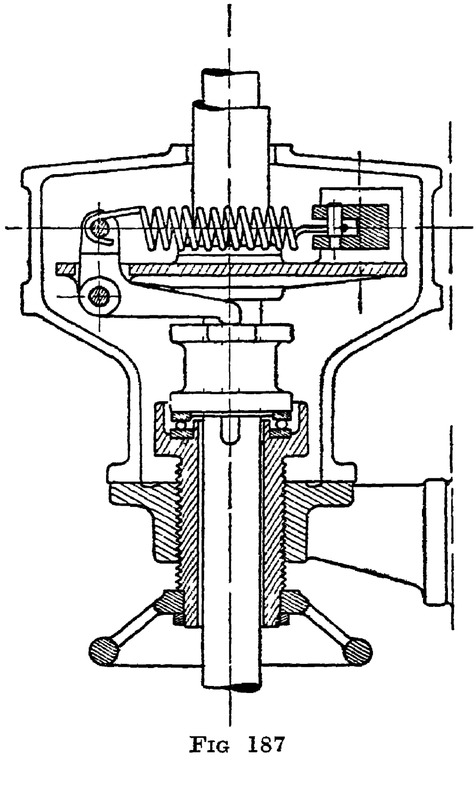

FIG 187

Variation of speed during the running of the engine is readily secured by transferring a part of the controlling force to an auxiliary spring the tension of which can be varied by mechanism provided for the purpose as shewn in Fig 186 or by varying the tension of the main spring itself as in Fig 187

Some points to be observed in governor design are —

(1) Weights as heavy as possible to give power and consequently render the effects of friction negligible

(2) Springs to be readily adjustable
(3) Small pin links etc to be as substantial as conveniently possible
(4) Friction to be reduced to a minimum by ball bearings
(5) Joints other than ball bearings to be bushed and provided with well hardened pins
(6) Lubrication both as regards supply of lubricant to the working parts and systematic disposal of the surplus to be considered carefully

The general disposition of the governor and fuel pump with respect to the framework of the engine is shewn in Figs 188

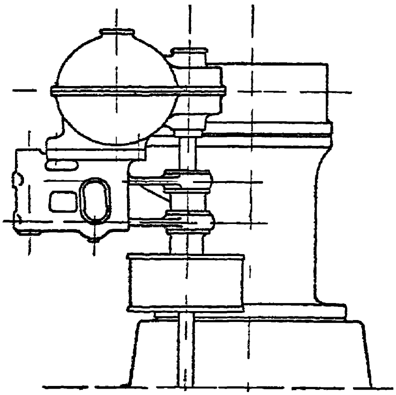

Fig 188

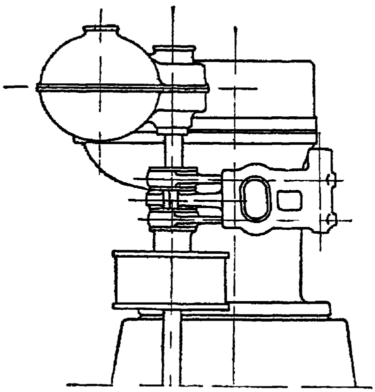

Fig 189

189 and 190 in three cases In Fig 188 the governor is of the angular movement type described above and illustrated in Fig 184 driven off the vertical shaft from which the cam shaft receives its motion The fuel pump is of the horizontal plunger type receiving its motion from eccentrics mounted on the same vertical shaft The fuel pump body is supported by a facing on the lower side of the case containing the upper spiral gears

In the arrangement shewn in Fig 189 the fuel pump is attached to the cylinder jacket but in other respects the details are similar to those of Fig 188

The governor shewn in Fig 190 is of the more usual type characterised by a sleeve which is mounted on a feather and which rises as the engine s speed increases The pump is of the vertical multi plunger type and regulation is effected by

rotation of an eccentric shaft on which are hinged the levers which operate the auxiliary plungers

The arrangements described briefly cover the bulk of the

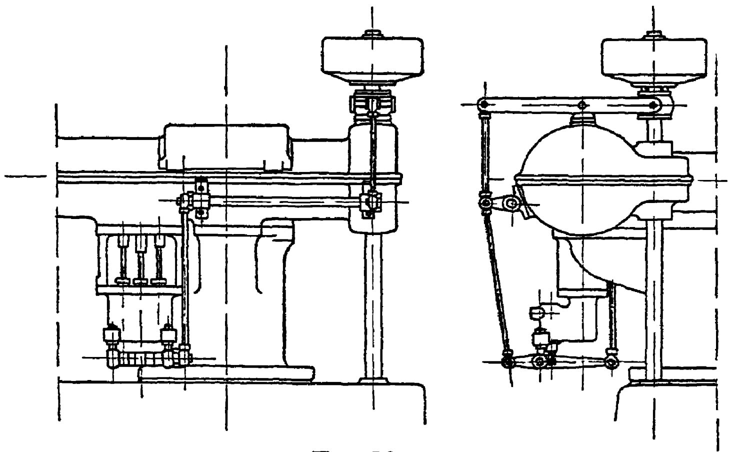

FIG 190

fuel pump and governor mechanisms found in practice but mention must be made of some modern refinements which are coming increasingly to the fore

(1) Control of fuel valve opening At light loads the duration of opening of the fuel valve is greater than necessary if uncontrolled and the instant of opening which is most favourable for full load running is inclined to be late for light load running At least one firm has attacked this problem of governor control of the fuel valve operating mechanism

(2) Blast pressure control This question is closely allied with (1) as a shortened opening period would lead to increase of the blast pressure if the latter were not corrected Apart from this the blast pressure has in any case to be altered in accordance with the load (unless cylinders are cut out of operation) if good combustion is to be secured at all loads including no load The blast pressure is placed under the control of the governor by means of a throttle slide on the compressor suction

(3) Pilot ignition This refers to cases where exceptionally refractory oils are being used which require for their

combustion a preliminary charge of a lighter oil such as Texas oil which is deposited in the pulveriser in advance of the main charge by a small auxiliary pump provided for the purpose The necessity for this device appears likely to be obviated by improvements in fuel valves and the pulverisers in particular

Fuel Injection Valves —It now remains to deal with the valve by means of which the fuel is injected into the combustion space and to which oil is delivered by the fuel pump for this purpose These may be broadly classified as the open and closed types respectively and as the former form a relatively small class at present it is convenient to dispose of them first

Open Type Fuel Valves —Fig 191 is a diagrammatic view of such a valve omitting all detail not required to illustrate

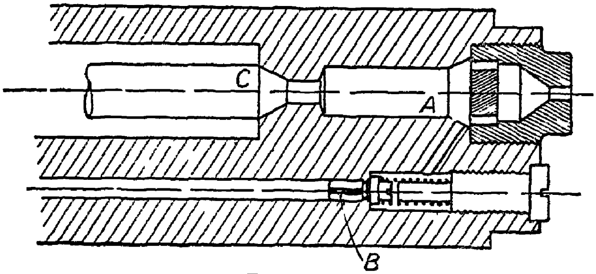

FIG 191

the bare principle Oil is delivered to the space A past the non return valve B by means of the fuel pump and this type of fuel valve derives its name from the fact that the space A is in constant communication with the interior of the cylinder It is to be noticed that the fuel pump is not required to deliver against the pressure of the blast air as the latter is restrained by valve C The latter is opened by appropriate gear at the predetermined instant for injection and carries with it the fuel oil contained in the space A The action appears to be highly efficient in pulverising effect and excellent fuel consumptions have been reported for engines in which these valves have been fitted This type of fuel valve appears to have been devised in the first instance for use in horizontal engines in which it was anticipated that the more usual type of fuel valve would be at a disadvantage A valve working on a somewhat similar principle has been tried from all accounts successfully on

vertical engines but has not yet to the author s knowledge become a standardised fitting

Closed Type of Flue Valves —In this type communication between the combustion space and the interior of the fuel valve only exists during the injection period when the flow is always in the same direction apart from such derangements as stuck valves or failure of the blast pressure

Fig 192 shews what may not improperly be called the Augsburg type of fuel valve Apart from the cast iron body the construction of which is sufficiently illustrated by the drawing the principal parts are —

(1) Needle valve A
(2) Spring B
(3) Stuffing box C
(4) Pulveriser D
(5) Flame plate E

The needle valve is usually made of special steel case hardened in way of the stuffing box to prevent cutting by the packing Accurate alignment of all parts of the needle is essential and readily secured by grinding between fixed centres The lower part of the needle is preferably reduced in diameter by a

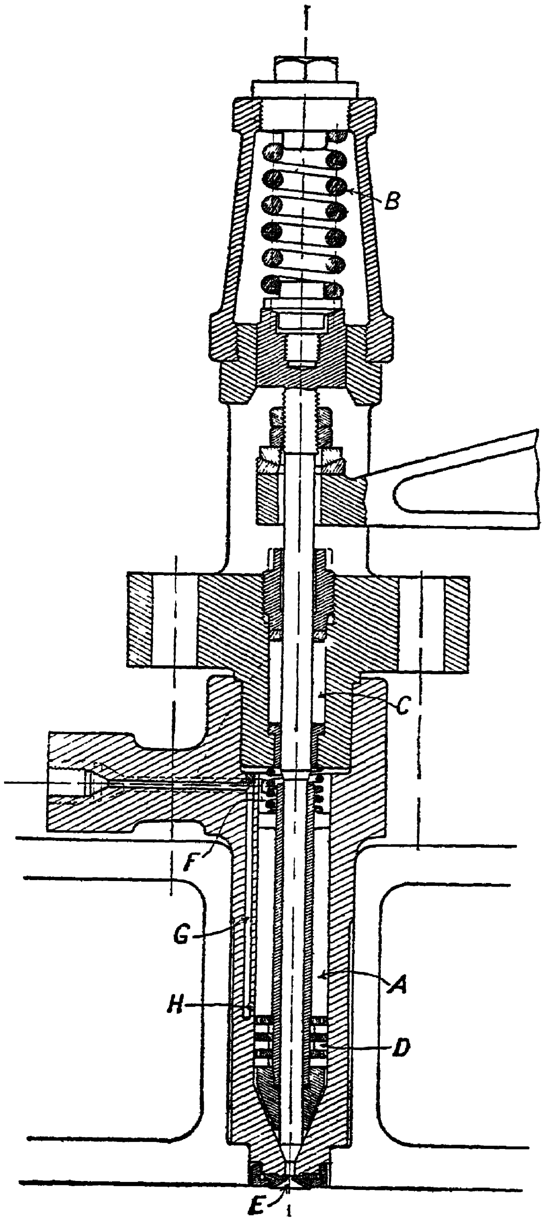

Fig 192

few thousandths as a certaın temperature gradıent exısts between the needle and the pulverıser tube whıch may lead to seızure ıf suffiicıent clearance ıs not allowed The tıp generally has an angle of about 40 degrees The needle sprıng ın addı tıon to returnıng the needle to ıts seat agaınst the pressure of the blast aır has to deal wıth the frıctıon of the stuffing box and may be figured out on the basıs of a pressure of 1500 lb per sq ın over the sectıonal area of the needle at the stuffing box The latter ıs usually provıded wıth a screwed gland

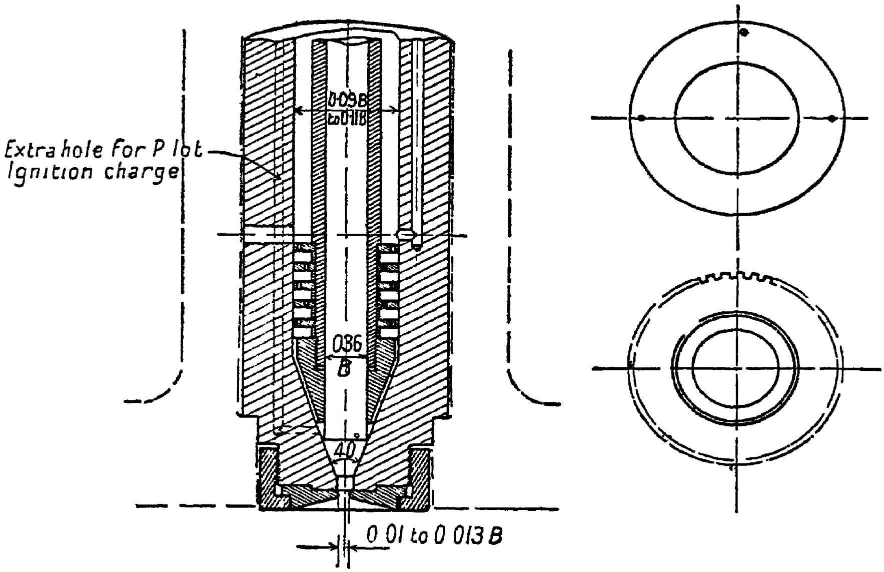

Fig 193

The pulverıser tube ıs held on ıts seat by a stıff sprıng and serves the double purpose of affoıdıng some support to the needle and retaınıng ın theır relatıve posıtıons the rıngs and the cone whıch play an ımportant part ın pulverısıng the fuel It will be clear from the figure that the pulverıser ıs surrounded by blast aır whıch enters at F The fuel ıs ıntroduced by means of a narrow hole G at a poınt H ımmedı ately above the top rıng If the poınt H ıs located too hıgh the oıl faıls to dıstrıbute ıtself evenly round the pulverıser rıngs and ıneffiicıent combustıon results

Fıg 193 shews the ınjectıon end of the pulverıser together

with the flame plate and nut to a larger scale The details shewn are those in most common use but are subject to variation in the practices of different manufacturers The proportions shewn are roughly indicative of good practice but it must be admitted that the rule of linear proportionality does not appear to be rational in this case Experience in this matter discloses two facts —

(1) That for a given engine there is a certain minimum diameter of pulveriser ring below which results are not satisfactory (about 9% of the cylinder bore)

(2) That as cylinders are increased in size it becomes increasingly difficult to obtain a high M I P

These suggest the following hypothesis —

That the best results are to be obtained when the depth of oil in the pulveriser before injection is a certain amount and the same for cylinders of all sizes If this is true then the area of the pulveriser ring should be in proportion to the cylinder volume This would lead to the diameter of pulveriser rings being made proportional to the cylinder bore raised to the power of 1 5 Such a rule has not been adopted and would probably lead to inconveniently large valves in the larger sizes of engines but the question would appear to offer some inducement to research A very large number of different types of pulveriser are in use and have been described in the technical press but it still remains to be proved that they are more efficient than the common variety shewn in Fig 193 A neat form of pulveriser tube which dispenses with the long narrow hole drilled in the fuel valve casting is shewn in Fig 194 from which it will be seen that the oil is led to an annular space A at the top of the tube whence it flows downwards to the pulveriser rings via a number of grooves in the surface of the pulveriser tube Holes B are provided to give passage for the blast air

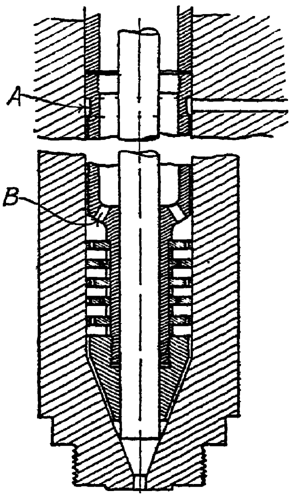

Fig 194

Swedish Type of Fuel Valve —Fig 195 shews the construction of this type of valve which has also been widely adopted and which is characterised by the fact that

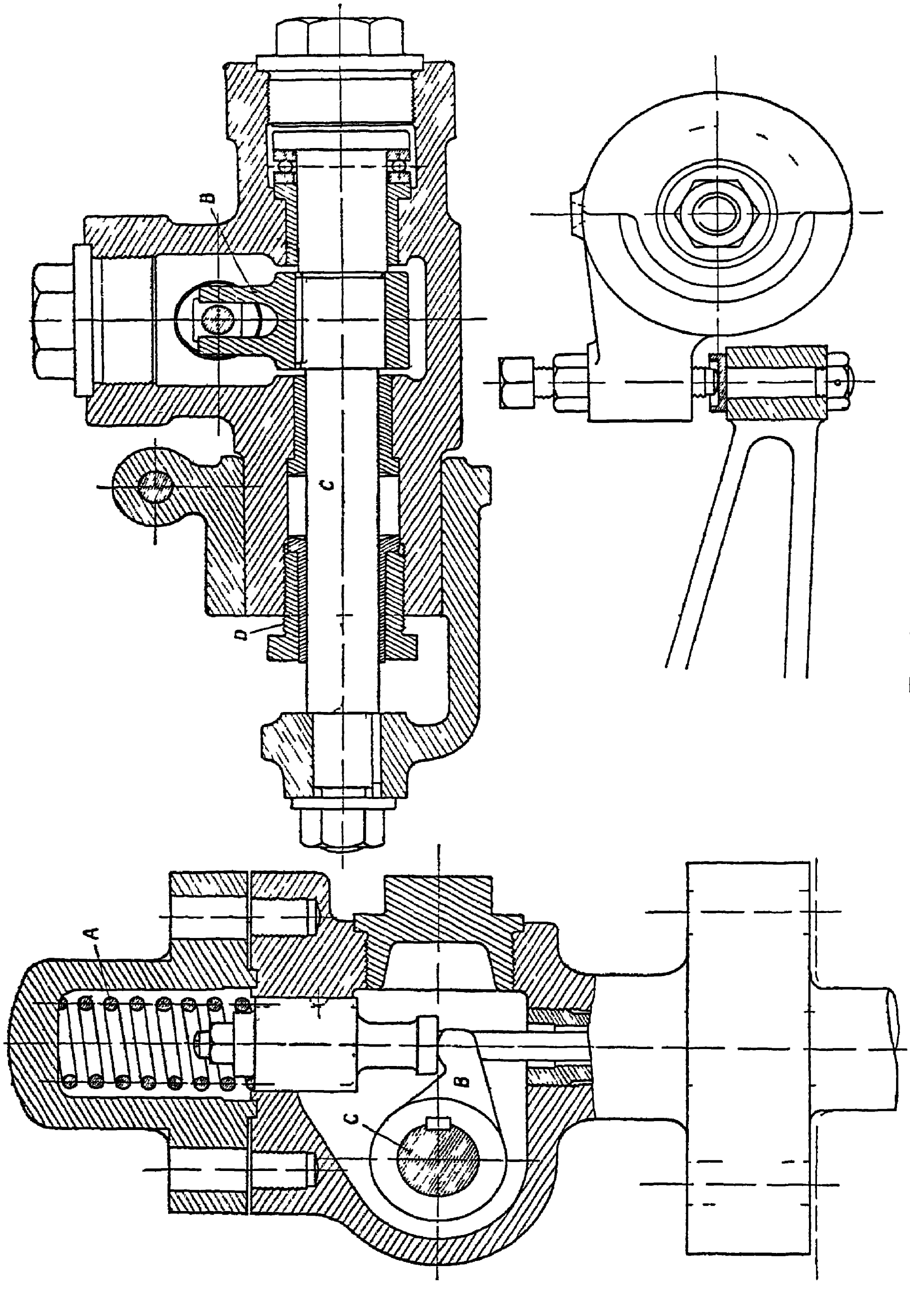

Fig 195

the needle is completely enclosed within the casing and is subject on all sides except the extreme tip to the pressure of the blast air On this account the spring A does not require to be as strong as that of an Augsburg type of valve of the same size The needle is lifted in working by the lever B attached to a cross shaft C the end of which penetrates the casing through a stuffing box D The mechanical means by which end thrust on the cross shaft and bending actions on the overhung end due to the pressure on the external lever are dealt with will be clear from the figure without further explanation The use of this type of valve appears to be limited at present to those designs in which the requirements of other parts of the valve gear necessitate the fuel valve operating lever being arranged off the centre line of the cylinder cover

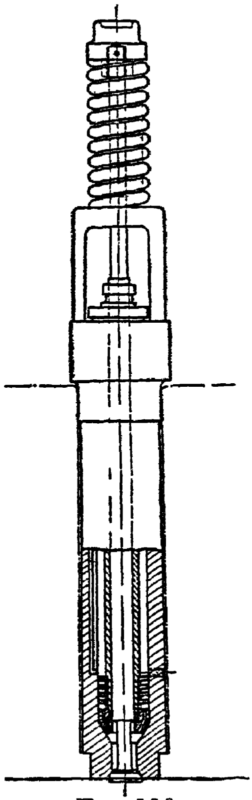

Fig 196

Burmeister Fuel Valve —The construction of this valve is shewn diagrammatically in Fig 196 and its outstanding features are the use of a mushroom valve the extreme simplicity of the whole arrangement and the fact that the valve is opened by a downward movement The latter is a particularly valuable feature as it secures uniformity of valve gear and ease of withdrawal

The four classes of fuel valve described above include as members most of the blast air fuel valves used on Diesel Engines at present Each type has its advantages but no one of them can be said to hold the field Something similar might be said for the enormous variety of pulverisers patented and in actual use It seems doubtful if any of these can claim outstanding efficiency When pilot injection of a less refractory oil is used to facilitate the use of tar oil as fuel an additional hole has to be drilled in the fuel valve as shewn dotted in Fig 193 The question of burning tar oil is still in the experimental stage in this country but the results so far obtained hold out hopes that it will be possible to dispense with pilot ignition in favour of special arrangements of a simpler character in connection with the fuel valve details

Some of the arrangements by means of which fuel valves are operated are shewn in Figs 197 198 199 and 200 The long lever which is a feature of all these schemes is usually of cast steel and should be of stiff construction The fulcrum on which the lever hinges is common to the levers which operate

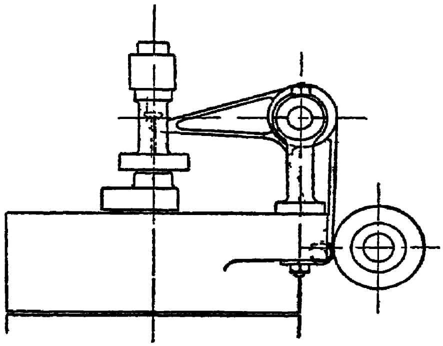

Fig 197

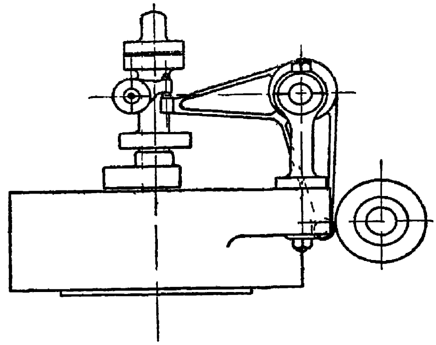

Fig 198

the other valves viz air and exhaust valves in the case of four stroke engines and scavenge valves in the case of two stroke engines and starting valves in both cases With land engines and many marine engines it is usual to mount the fuel and starting valve levers on eccentric bushes mounted on the

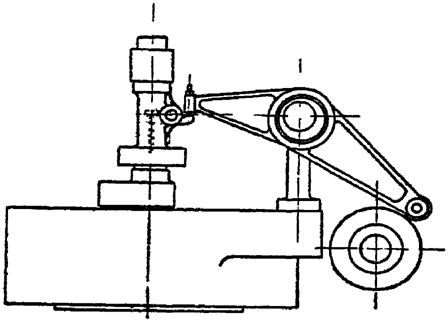

Fig 199

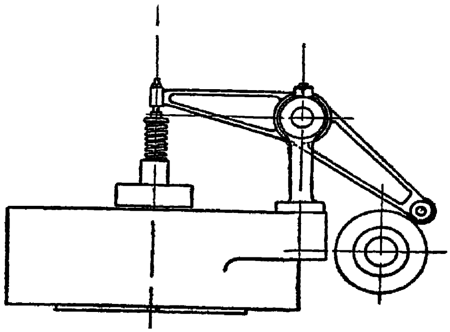

Fig 200

fulcrum shaft at such angles that the operation of putting the starting valve into gear automatically puts the fuel valve out of gear and vice versa This is considered in detail in Chapter XIV

The use of the needle type of fuel valve in conjunction with a single lever necessitates the latter being so disposed that its

roller is rendered more or less inaccessible by the cam shaft particularly if the latter runs in a trough (see Fig 198) The difficulty may be got over by providing a small intermediate lever as shewn in Fig 199 to reverse the direction of motion In spite of the objections which have been raised against this arrangement it appears to be satisfactory in practice

Design of Fuel Valves—An approximate rule for the internal diameter of the body has already been given being the same as the diameter of the pulveriser rings The thickness of the walls (cast iron) may be from a third in large valves to a half in the case of small valves of the internal diameter If the valve is of the Swedish type this thickness will be approximately constant throughout the body of the valve except in the neighbourhood of flanges etc If of the Ausgburg type those parts of the body not subject to pressure may be a little thinner In all cases a good rigid job should be aimed at as lack of alignment leads to sticking of the valve The pulveriser tube is made of steel and the details such as rings and cones of steel or cast iron The Swedish type of valve requires special care to be devoted to the design of the cross shaft and its fittings in order to obtain freedom under load adequate bearing surface and accessibility of the stuffing box As regards the valve as a whole the designer should aim at shapely solidity and avoid flimsiness of detail

With the Augsburg type of valve (Fig 192) the load necessary to lift the needle is the spring load less the product of the blast pressure and the area of the needle at the stuffing box (approx) plus the gland friction With the Swedish type (Fig 195) the load may be taken as approximately equal to the spring pressure plus the product of the blast pressure and the area of the needle at its seat This load evidently induces bending and twisting actions which the cross shaft should be proportioned to carry with a low stress The weakest section is generally at the reduced diameter to which the external lever is keyed The key itself should be amply proportioned and is preferably made of tool steel The ball thrust must be proportioned to the load obtained by the product of the maximum blast pressure into the sectional area of the cross shaft at the stuffing box The flame plate is of nickel steel and the diameter of the hole is usually about 1 % of the cylinder bore but the best size for any particular case must be found by experiment The flame plate nut may be of steel or bronze

secured to the fuel valve body by a fine thread and provided with flats to accommodate a spanner

The main points in the design of fuel valves may be summarised as follows —

(1) Rigidity and alignment of casing
(2) Alignment of the needle and its guide
(3) Freedom of all working parts
(4) Sturdy proportions for all small details

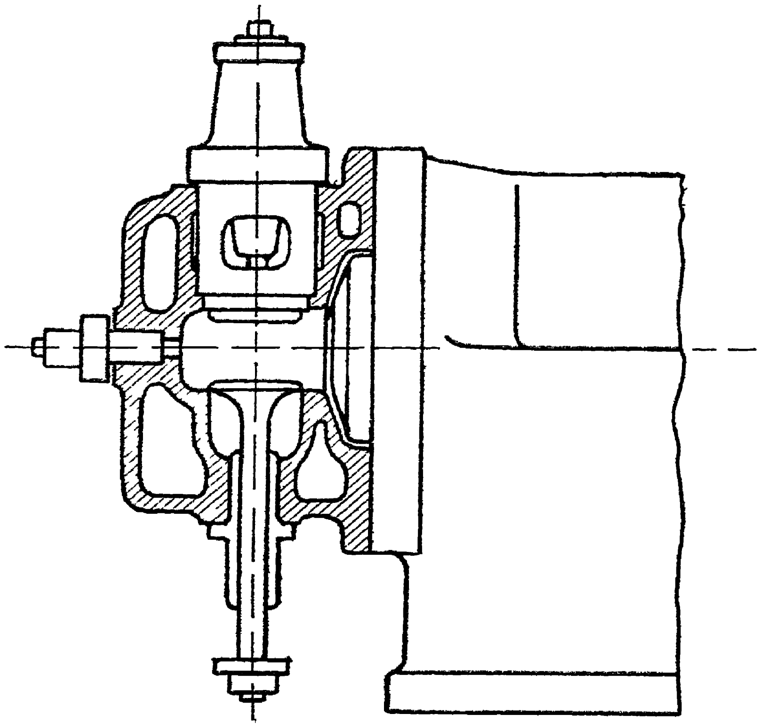

Fig 201

Mechanical Injection —The various kinds of Diesel type engines known as solid injection airless injection or cold starting heavy oil engines which share the characteristic Diesel feature of spontaneous ignition but differ from true Diesel Engines inasmuch as the fuel is injected by mechanical means instead of by an air blast owe their origin to two distinct lines of development

In one line of development firms already experienced in the manufacture of the true Diesel have sought to eliminate the air compressor for injection purposes with a view to simplification and reduction of cost In these cases the traditional arrangement of valves and a concave piston top is usually retained but the compression is usually reduced to 380 to 400 lb per square inch and the indicator card reveals a certain amount of combustion at constant volume so that the maximum pressure attains a value of 500 to 650 lb /in The use of compressed air for starting purposes is retained

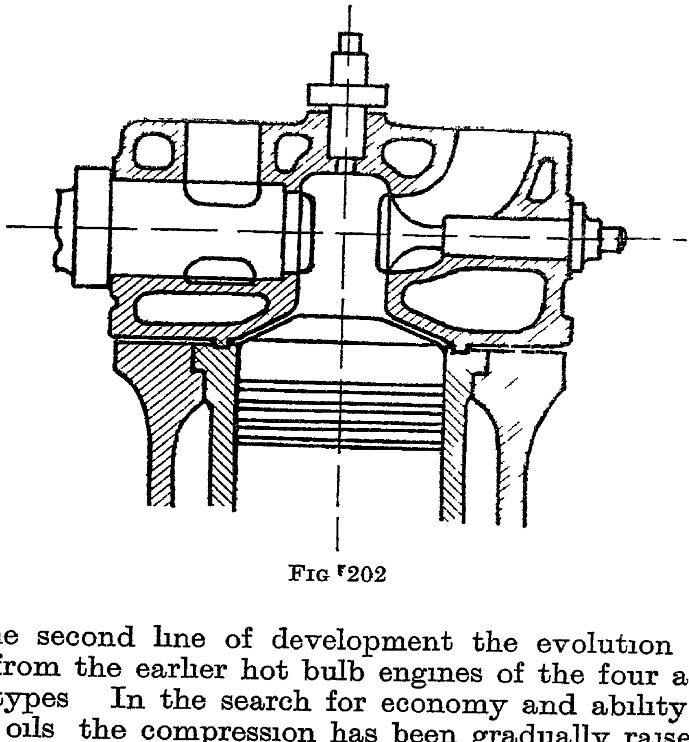

Fig 202

In the second line of development the evolution can be traced from the earlier hot bulb engines of the four and two stroke types In the search for economy and ability to use heavier oils the compression has been gradually raised until at last the use of externally heated surfaces can be dispensed with At this stage the compression pressure and the indicator diagram generally correspond with those attained as indicated above Such engines were usually of the horizontal type in the case of four stroke engines with the inlet valve arranged over the exhaust valve and the combustion chamber in the form of a compact pocket (Fig 201)

There are now a number of makes of such engines both of horizontal and vertical types on the market The character istic arrangement of combustion chamber and valves is retained in the vertical engines also in some cases (Fig 202)

The question whether the adoption of mechanical injection actually does achieve simplification and reduction of cost as compared with air blast injection is a very debatable one which we shall not attempt to discuss fully here Given equal sizes of cylinders in the two cases it seems clear that the mechanical injection engine must be cheaper to build even after allowance is made for higher maximum cylinder pressures

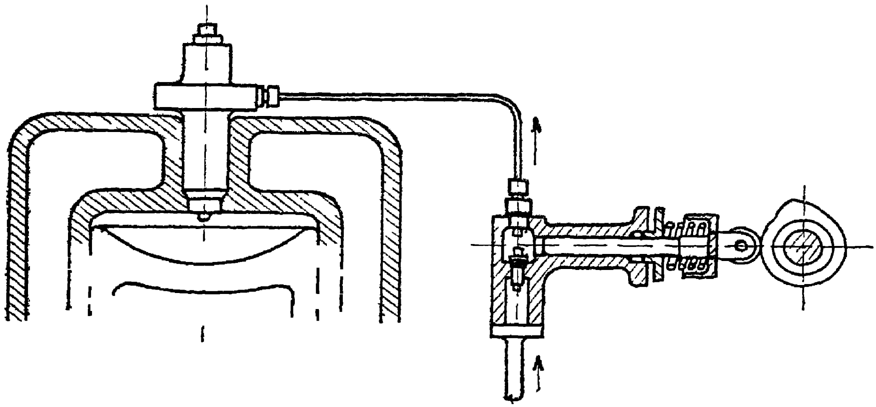

FIG 203

On the other hand experience shows that the air blast engine can achieve the higher brake mean pressure in spite of its mechanical efficiency being lower by about 5 or 6% It then becomes a question which engine can safely maintain the higher brake mean pressure under service conditions and this experi ence alone can decide

The Combustion Chamber —The ability of the air blast engine to carry a higher brake mean pressure (and therefore *a fortiori* a higher M I P) than the mechanical injection is traceable to two causes

(1) The supercharging effect of the blast air which increases the available oxygen by about 5 to 10%

(2) The action of the air blast in promoting rapid com bustion (turbulence)

In the mechanical injection engine the turbulence is

dependent upon the air speed through the inlet valve (about 150 ft /sec) and the effect produced by the piston displacing the charge into a pocket shaped combustion space when this construction is adopted

Any attempt to increase the air inlet speed unduly would react unfavourably on the pumping loss and reduce the charge weight In order to obtain satisfactory combustion with the limited turbulence available it is necessary to get the fuel in quickly and early in order to gain time for combustion and to attain a high temperature as quickly as possible This means

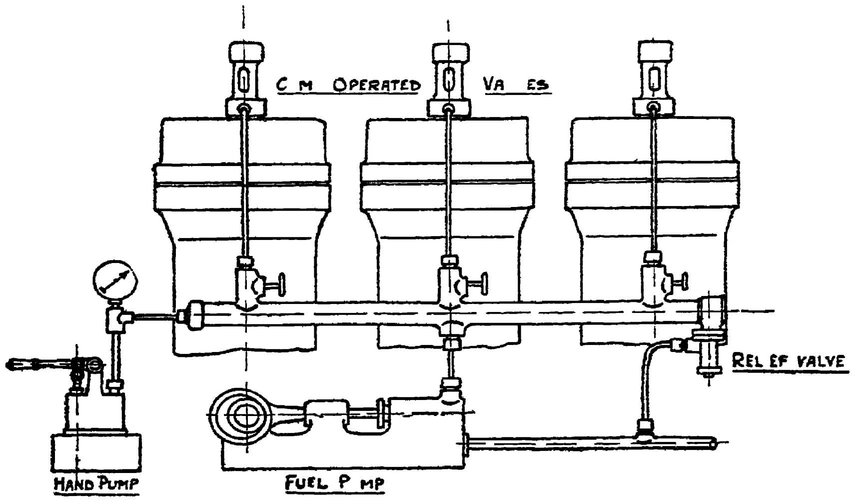

FIG 204

a period of combustion at constant volume with consequent rise of pressure at the dead centre As very high maximum pressures are undesirable the compression pressure is accordingly lowered to the lowest point which will secure certain ignition under the circumstances contemplated A compression pressure of about 350 lb /in ^{2} seems to be about the lower limit in practice unless preheating of the jackets can be resorted to When starting has to be effected with cold jackets on heavy oils 400 lb /in ^{2} appears to be desirable

Mechanical Injection Systems —The existing systems can be broadly divided into two classes In the first class a fuel pump is provided for each cylinder and the injection period corresponds to the delivery period of the pump which is

frequently cam operated (Fig 203) The fuel valve is entirely automatic in action and consists either of a simple capillary orifice or more usually of a spring loaded needle valve which is lifted by the pressure of the fuel acting on an unbalanced area

In the second class the fuel valves are mechanically operated in much the same manner as with the air blast system Fuel is supplied to them from a common main or reservoir main tained at a sensibly constant pressure by a pump of one or more plungers (Fig 204) This system is sometimes referred to as the rail system The pressure in the rail or reservoir is maintained practically constant at a value of 2000 to 8000 lb /in ^{2} by providing sufficient volume capacity to absorb the fluctuations in demand and supply by the elasticity of the fuel itself In earlier designs special arrangements of accumulators or resilient members were provided but it is easy to shew that these are unnecessary

Suppose that the permissible pressure fluctuation is 4000 to 4500 lb /in ^{2} whilst a charge of 1 in ^{3} of fuel is being pumped into the main According to Kaye and Laby the modulus of compressibility of petroleum is —

$$C=\frac{1}{V}\frac{\delta V}{\delta p}=69\ 5\times 10^{6}\text{ per atmosphere where}\begin{cases}V=\text{volume}\\ p=\text{pressure}\end{cases}$$

so the required capacity is —

$$1\text{ cub in}\times\frac{10^6}{69\ 5}\times\frac{14\ 7}{500}=423\text{ cub in}$$

apart from the yielding of the tube wall which is in general small compared with the yielding of the fluid itself

Fuel Pumps for Mechanical Injection —With the rail system the fuel pumps are most conveniently driven by eccentrics and the construction of the pump may follow very closely the designs already referred to in connection with the air blast system with the following differences —

(1) The high pressure used viz 2000 to 8000 lb /in ^{2} demands a stronger construction of eccentric and plunger connections and more rigid connection between the pump body and the eccentric shaft bearings

(2) The pump body is preferably made of forged steel or bronze as cast iron is liable to be too porous

(3) A relief valve and by pass are provided in order to maintain a constant pressure in the delivery main
(4) The full stroke of the plunger can be utilised for delivery allowance being made for overload and slip
(5) The clearance volume of the pump should be small to reduce slip by re expansion of the compressed oil as in an air compressor
(6) The slightest air lock must be avoided
(7) The delivery fittings must be very substantial to stand strenuous tightening up The same applies to the plunger gland unless a lapped plunger working in a plain sleeve is adopted

In those systems in which an individual pump controls the injection to each cylinder the delivery stroke must only occupy about 30 crank shaft degrees at full load and less at light loads This is usually arranged by one of two alternative methods In the first the plunger is driven outwards on the delivery stroke by means of a cam with a fairly sharp lift of the desired duration and returned on the suction stroke by a spring the returning side of the cam being fairly gradual The volume delivered is varied in accordance with the requirements of the load by some such device as a wedge piece interposed between the plunger and the cam follower the position of the wedge piece being under the control of the governor

At full load the thick end of the wedge is interposed and the full lift of the cam is utilised at light load a thinner part of the cam is interposed and since the return stroke of the plunger under the influence of the spring is limited by a stop there is clearance between the cam follower and the wedge and consequently a shorter stroke of the plunger This arrangement and others equivalent to it suffer from the defect that the point at which delivery starts is retarded as load is reduced

In the second method this objection is overcome The plunger operates with a constant stroke determined by a cam or eccentric but the effective length of the stroke is shortened on light load by the forcing open of a so called spill valve which allows the escape of fuel from the pump barrel to the suction side of the pump The point at which the spill valve lifts is varied in accordance with the load by means essentially similar to those employed in the true Diesel fuel pump for regulating the point of seating of the suction valve (Fig 205)

With this kind of pump the instant of injection remains constant at all loads

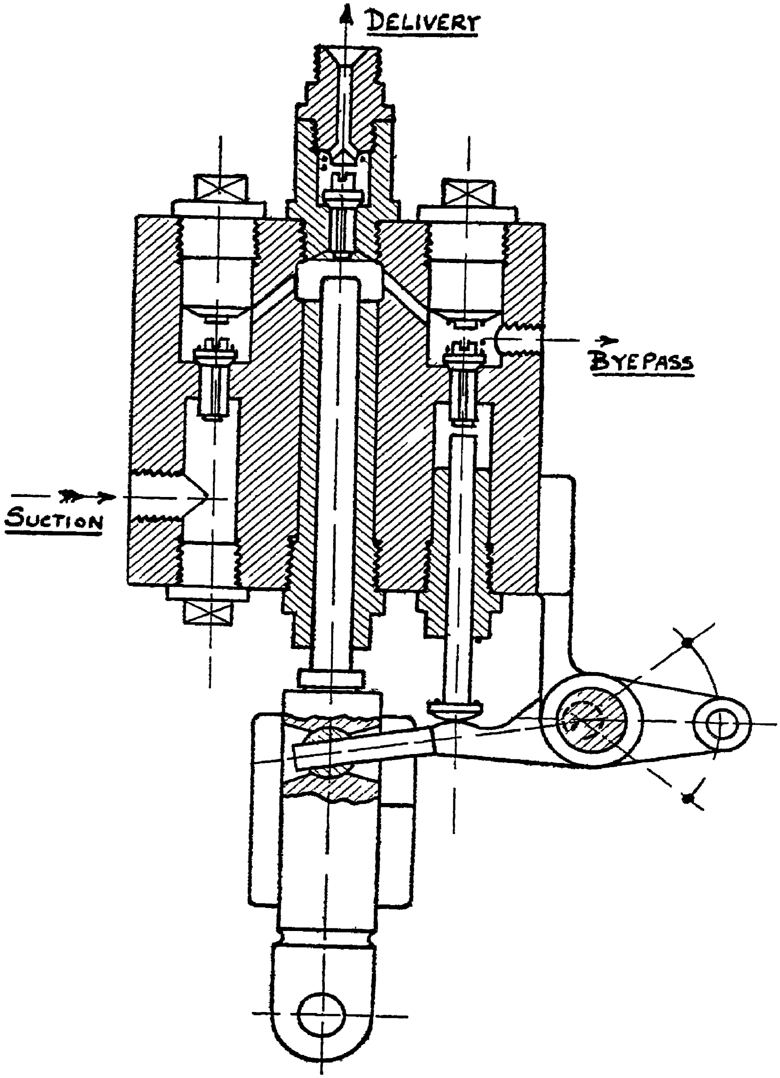

FIG 205

With these pumps it is necessary to arrange the governing mechanism whether spill valve gear or wedge mechanism or equivalent in such a way that the operating forces do not

appreciably react on the governor and are not subject to undue wear Simplicity of adjustment should also be considered

Fuel Valves —The mechanically operated fuel valves associated with the rail system have been evolved from the air blast types of valve and designs corresponding to the Augsburg and Swedish types have been used Apart from the elimination of the blast air arrangements the essential differ ences are as follows —

(1) The interior of the valve is filled with fuel oil instead of air and plugs are arranged to vent air locks if neces sary

(2) The valve body is made of forged steel or bronze to withstand the high pressure

(3) The spring pressure per unit of needle area is greater (in the Augsburg form) for the same reason

(4) Drains are provided to carry away any gland leakage

(5) Pulverisers are not required as the oil is broken up on emerging at high speed from the nozzle orifices

(6) The needle is carried down as near to the nozzle orifices as possible to pre vent after drip i e the slow emergence of oil from the orifices after the valve has closed (Fig 206)

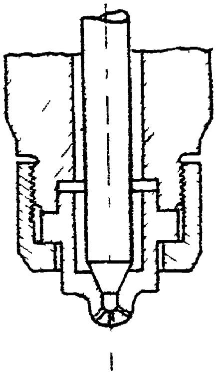

FIG 206

(7) The nozzle or flame plate usually con tains a plurality of minute holes of the order 20/000 in diameter or a single hole of larger size

(8) A filter capable of stopping particles of grit of a diameter less than that of the nozzle orifices is put in circuit adjacent to the valve itself or even incorporated in the body of the latter This is very desirable if not essential as scale and grit in the connecting pipes are difficult to avoid

(9) Means must be provided to vary the period of opening of the valve in accordance with the load This can be achieved by eccentric mounting of the valve operating lever or equivalent means whereby roller clearance is increased This in general retards the point at which injection starts as load is reduced but

this defect can be overcome by arranging that the movement of the cam roller during the increase of roller clearance shall follow the cam profile or even interfere with it thus obtaining an advance of the point of injection

Fuel valves of the automatic type may consist of a simple plug with a union at the outer end and a nozzle plate at the combustion end connected by a hole of small diameter with or without a non return valve near the nozzle end It seems to be pretty well agreed however that this type of valve is liable to after drip and that the best economies cannot be obtained with it

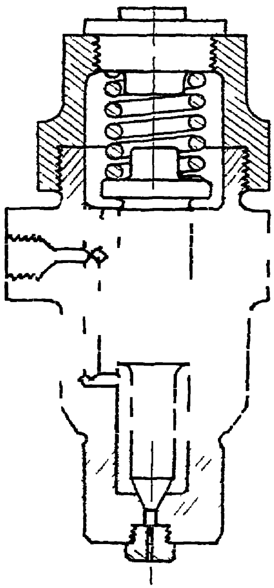

FIG 207

The preferred type contains a spring loaded needle valve seating close to the nozzle The outer end of the needle passes through a bushed hole in the valve body thus providing an unbalanced area on which the fuel pressure acts in lifting the needle against the spring pressure during the pump delivery period (Fig 207) The spring tension per unit of needle area at the bushed hole determines the fuel pressure at the point of injection

If the needle lifts against a stop the mean pressure during injection may be considerably greater than this if the nozzle orifices are sufficiently fine If the latter are relatively coarse the needle lift will be very slight and the injection pressure will be sensibly constant

Fuel Valve Nozzles —A great deal of experimental work has been done by the various makers of mechanical injection engines to secure the most favourable type of fuel sprayers the efficiency of the combustion process depends very largely on the degree of pulverisation and the character of the distribution of the fuel Some of this work is referred to in the list of references at the end of the chapter It seems to be established that the degree of pulverisation should strike a happy medium between the extremes of —

(a) Too coarse pulverisation leading to slow combustion

and carbonisation due to liquid oil striking the piston or otherwise

(b) Too fine pulverisation preventing the spray from adequately penetrating the charge and leading to undue concentration of oil in the neighbourhood of the injector with the result that combustion is incom plete and smoke is formed

In an ideal state of affairs the fuel in a fine state of division would be evenly distributed throughout the hot central core of the combustion chamber When the latter consists of an elongated pocket or egg shaped chamber the ideal programme can be approximately realised by the use of a single orifice nozzle

If on the other hand the combustion chamber is wide and shallow a plurality of orifices is usually employed

The size of orifice required may be determined approxi mately as in the following example —

I H P per cylinder (4 stroke cycle)	100
R P M	300
Fuel consumption per I H P hr in lb	0 35
Density of fuel in lb per ft ^{3}	55
Injection pressure lb /sq in (above ignition pressure)	4000
period in crank shaft degrees	30

Volume of oil injected per working stroke

$$=\frac{100\times 0\ 35}{60\ \times 150\times 55}\text{ cubic feet}$$

Duration of injection period

$$=\frac{60\times 30}{300\times 360}=\frac{1}{60}\text{ sec}$$

Rate of discharge during injection period

$$=\frac{60\times 100\times 0\ 35}{60\times 150\times 55}=0\ 00425\text{ ft }^3\text{ per sec}$$

Head corresponding to 4000 lb /in ^{2}

$$=\frac{4000\times 144}{55}=10\ 520\text{ ft}$$

Apparent velocity allowing coeff of discharge $=0\ 6$

$$=0\ 6\sqrt{2\ g\times 10\ 500}=495\text{ ft /sec}$$

$$\text{Area required} = 0\ 00425 - 495\ \text{ft}^2 = 0\ 00086\ \text{in}^2$$

$$\text{Required dia for single orifice} = \sqrt{\frac{4}{\pi \times 860}} = 39/1000$$

$$,\qquad 6\ \text{equal orifices} = 39/1000 - \sqrt{6} = 16/1000$$

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CHAPTER XII

AIR AND EXHAUST SYSTEM

Four Stroke Engines—So far as four stroke engines are concerned the parts included in this system are —

The air suction pipe
Air suction valve
Exhaust valve
Exhaust piping
Silencer

In the neighbourhood of cement works or other sources of grit a suction filter is sometimes added with a view to preventing foreign matter from reaching the interior of the cylinder along with the indrawn charges of air

Air Suction Pipes—The usual form of suction pipe for both land and marine engines is shewn in Fig 208 and is intended to serve the double purpose of muffling the sound caused by the rush of air past the inlet valve on the suction stroke and also to prevent in some measure the ingress of dust To be of real service in either capacity the slots should not exceed about 40/1000 of an inch in width and in use must be kept clear of dirt otherwise the volumetric efficiency and consequently the power of the engine will be seriously impaired To be reasonably effective as a silencer the slots should cease within a foot or so of the point where the pipe joins the cylinder cover To prevent throttling the aggregate area through the slots is usually made about 1 5 to twice the clear area of the pipe Assuming the slots are spaced $\frac{3}{8}$ apart the slotted length of the pipe works out to about $3\frac{1}{2}$ to $4\frac{1}{2}$ times the bore In very

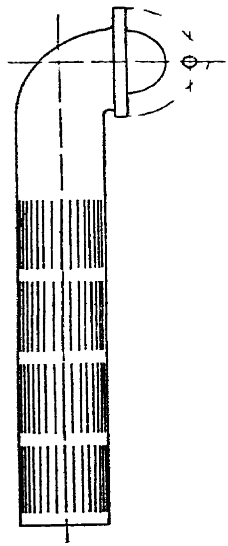


FIG 208

expert foundries these slots are formed in the casting (cast iron) Where the slots have to be milled aluminium is a convenient and suitable material for this purpose Welded tubes of sheet iron are sometimes used but are liable to become dented and unsightly

An efficient and durable arrangement shewn in Fig 209 consists of a common collecting pipe in communication with all the cylinders and ending in a trumpet shaped piece which is very effective in muffling the sounds of suction The trumpet fitting is similar to a cornet mute and consists of internal and external members so arranged that the space between them has a sectional area increasing outwards whilst the distance between the two members diminishes

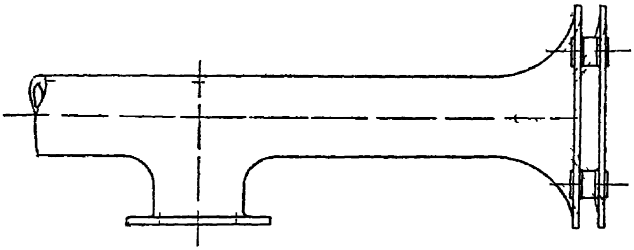


FIG 209

The noise question is most effectively disposed of by carrying the suction pipe outside the building or engine room in dusty localities If an air filter be added the arrangement is ideal

Where a separate suction pipe as in Fig 208 is fitted to each cylinder the bore is usually made equal to that of the suction valve or slightly larger Where a common suction pipe is provided the bore may be anything up to about double this size according to the number of cylinders to be supplied and the length of the pipe

Suction Valves —A typical suction valve is shewn in Fig 210 The valve proper may be of carbon or nickel steel in the form of a drop forging The upper end is guided by a little piston which also takes the thrust of the spring In the example shewn the guide piston is of chilled cast iron extended in the form of a nut and provided with a cup shaped cavity to accommodate the operating tappet The casing being in two parts it is necessary to finish each part on a mandril to ensure alignment in position It is not necessary at this point to describe

variations in the details of suction valves as the latter are usually made similar to the exhaust valves (which will be described in detail later) except for such special features as are necessary to deal with the heat effects to which the latter are subject

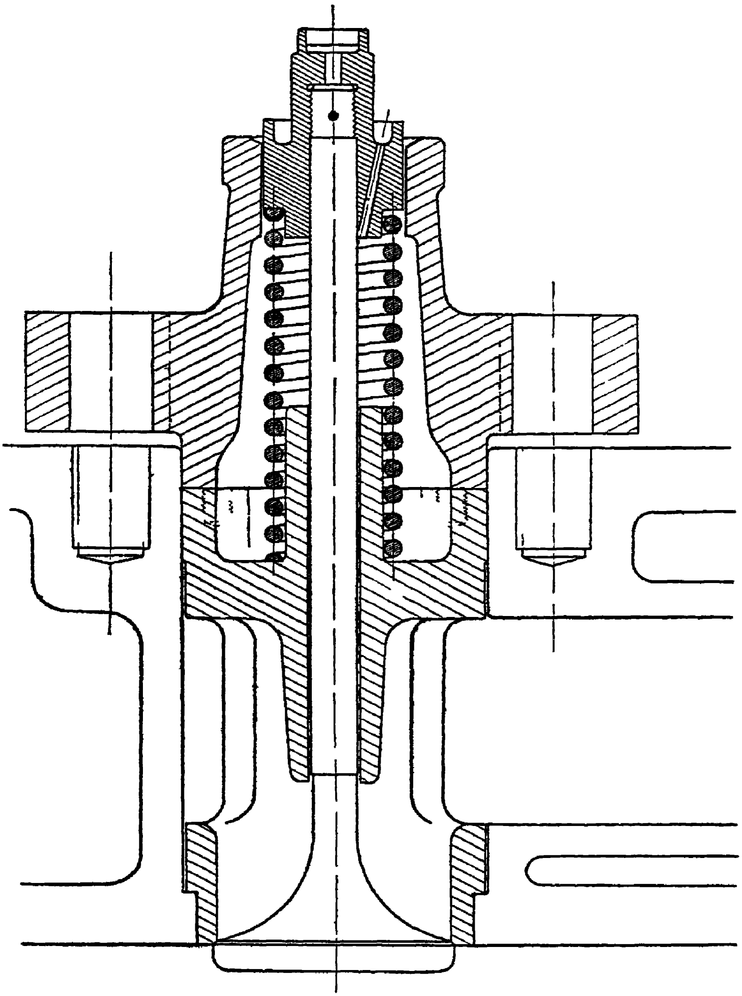


FIG 210

Dimensions of Air Suction Valves —The requisite diameter for an air suction valve may be regarded as determined by the mean vacuum allowable on the suction stroke Taking this to

be 0 6 lb below atmospheiic pressuie the theoretical mean velocity is found from Fig 13 to be about 280 feet per second and taking the mean coefficient of discharge to be 0 70 this gives a mean apparent velocity of 195 feet per second Now as regards the mean opening area of the valve if we assume a harmonic cam opening and closing exactly at the upper and lower dead centres respectively then the mean opening would be just half the maximum opening if the maximum lift is made $=\frac{1}{4}$ of the valve diameter Actually the valve is always arranged to open before top centre and close after top dead centre so that the mean area is usually more like 0 6 of the maximum area

Adopting this figure we may write —

$$195 = V = V_p \times \frac{0\ 785\ B^2}{0\ 65 \times 0\ 785\ d^2}$$

Where V = Apparent mean velocity of air ii feet per second
V_p = Mean piston speed in feet/seconds
B = Bore of cylinder
d = Diameter of suction valve

From which

$$\frac{d}{B} = \sqrt{1\ 54 \frac{V_p}{195}} = \sqrt{\frac{V_p}{127}}$$

Values of $\frac{d}{B}$ calculated fiom this formula for various piston speeds are given below and agree well with average practice

Piston speed in ft per min	500	600	700	800	900	1000	1100	1200
Ratio d—B	257	281	303	324	343	362	380	397

In the best practice the maximum lifts of the air and exhaust valves are frequently made as much as 0 28 of the valve dia meter as although the extra lift does not increase the maximum available area the mean area is increased and the opening at the dead centre is augmented without adopting an unduly long opening period or an awkward shape of cam

The air and exhaust valves are usually operated by the lever arrangement shewn in Fig 200 with reference to fuel valves

Exhaust Valves —In some small engines the exhaust valves are similar to and interchangeable with the air suction valves but with medium and large size engines owing to the larger dimensions of the exhaust valves and in consequence their slower rate of cooling by conduction to the surrounding media

special arrangements have to be made to conduct heat away from the valve seat which would otherwise become pitted and grooved in a short time

With cylinders up to about 16 in bore the trouble can be reduced to reasonable proportions by providing the valves with cast iron heads which are less liable to become pitted than those of nickel steel Ventilation of the casing by means of perforations is also found useful Some types of cast iron exhaust valve heads which have proved successful in practice are shewn in Fig 211 With cylinders much in excess of 16 in bore some form of water cooled casing is desirable and the simple arrangement shewn in Fig 212 has proved most

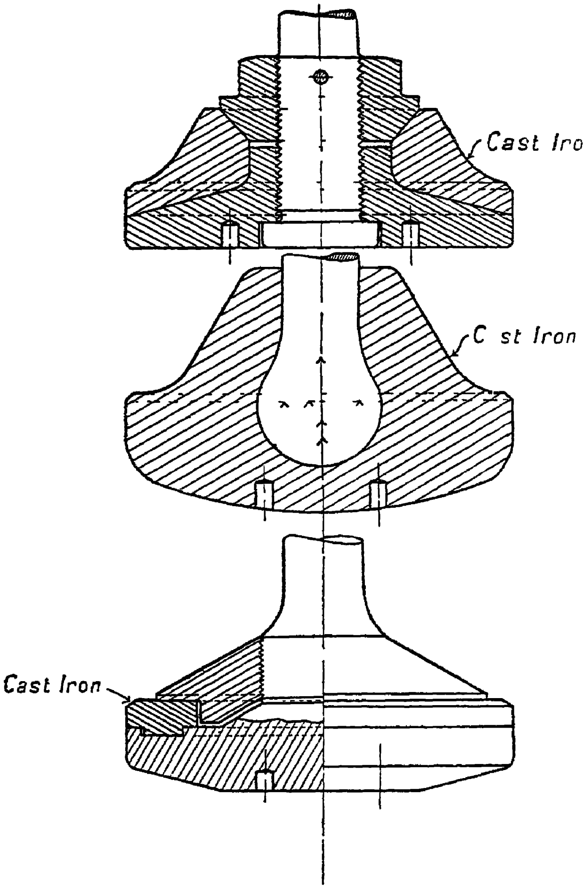


FIG 211

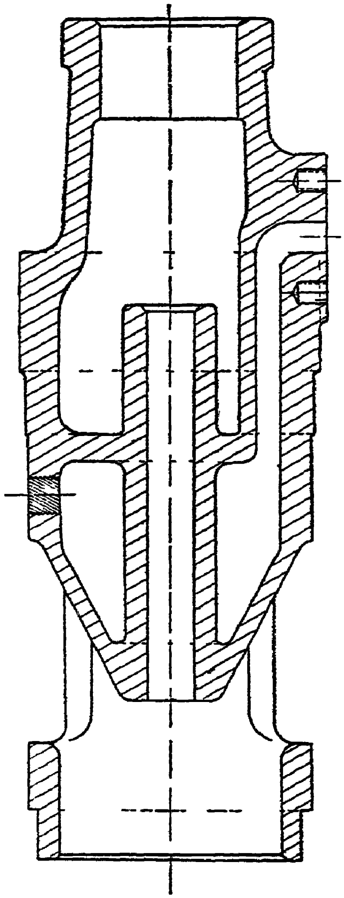


FIG 212

effective with the very largest cylinders without resorting to the expedient of cooling the valve itself by direct means It appears that the flow of heat from the seating to the water jacket is sufficient to keep the temperature of the former at a suitable low value so long as the mean indicated pressure does not exceed a reasonable figure dictated by other considerations

The valve spindle is liable to become stuck by carbonaceous deposit and should therefore be about 20/1000 slack in the casing Furthermore arrangements should be made to feed a little paraffin to this point from time to time

The upper or guide end requires some care in detailing

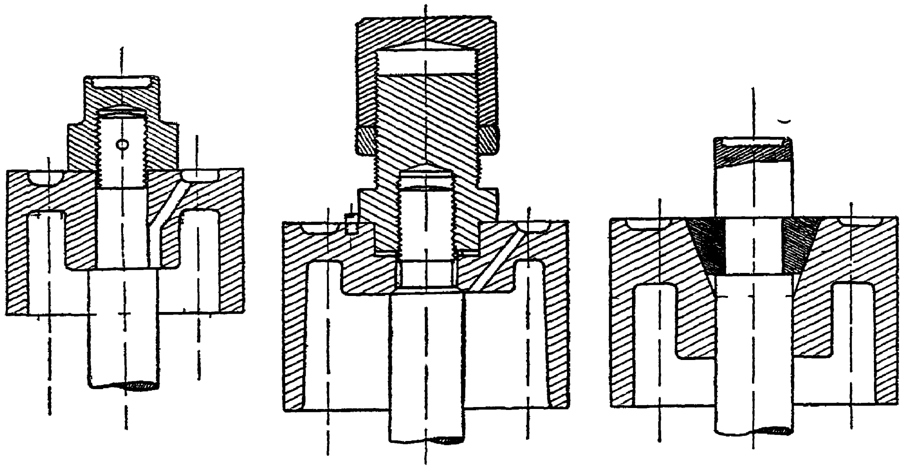


FIG 213

particularly in high speed engines to prevent the nut from slacking back or the threads becoming stripped The arrangement shewn in Fig 210 is simple and effective other arrangements are shewn in Fig 213 for slow speed engines

Valve Casings—Two types of valve casings have been illustrated in Figs 210 and 212 and further modifications are shewn in Fig 214 The valve seating is made in a separate piece which is readily replaceable when worn by an interchangeable spare After repeated regrindings a ridge is formed which has to be turned off a process more conveniently carried out on a light ring than on a whole casing If no loose seat is provided it becomes necessary in time to turn down the under surface of the casing flange in order to bring the valve seat

flush with the under side of the cover The port in the casing provided for the flow of exhaust gas should accurately coincide with the corresponding port in the cover and it is usual to provide an equal and opposite dummy port to preserve the symmetry of the casting and diminish chances of distortion due to expansion or growth The depth of the metal under the ports should not be reduced to too fine a limit otherwise there is danger of leakage on the compression and expansion strokes due to deflection or even cracking of the metal at this point The number of studs (two to four) used to secure the casing to the cylinder cover is purely a matter of convenience in arranging the gear on the cover or cylinder head Two studs properly proportioned are sufficient for the largest valves given an adequate depth of flange and good connection of the latter to the body

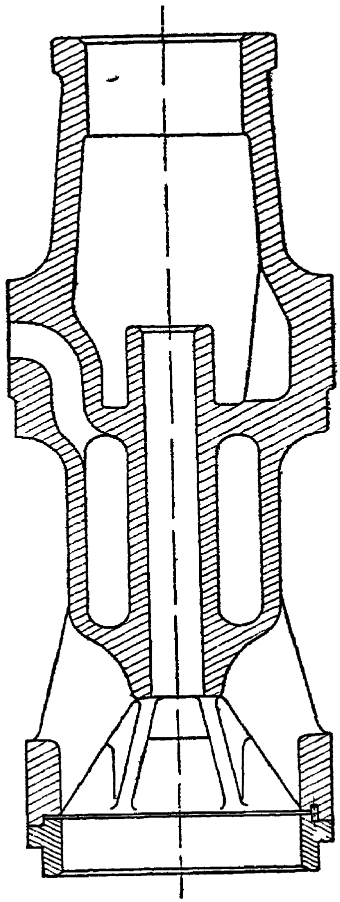


FIG 214

Exhaust Lifting Devices —Some form of exhaust lift for breaking compression is usually fitted to both land and marine engines With the former the use of such a device on shutting down the engine obviates the tendency of the engine to swing in the reverse direction to that for which it was designed just before stopping and also facilitates turning the engine by hand With marine engines some such device should come into operation automatically on reversing to prevent the possibility of compressing an unexhausted charge when motion is begun in the reverse direction An expansion stroke executed in the ahead direction becomes a compression stroke in the astern direction and vice versa Two hand devices are shewn in Fig 215 Type A consists of a substantial steel lever provided with a slotted end which normally keeps it in a vertical position clear of the gear To bring the lever into operation it

is lifted to the extent of the slot and allowed to fall forwards A projection on the lever then slips under an extension of the roller pin on the first occasion of the valve being lifted and prevents its return Type B consists of a link and screw by means of which the valve may be depressed during the running of the engine and which normally lies alongside the casing Fig 216 shews three arrangements suitable for marine engines In type A the valve is depressed by a pneumatic cylinder

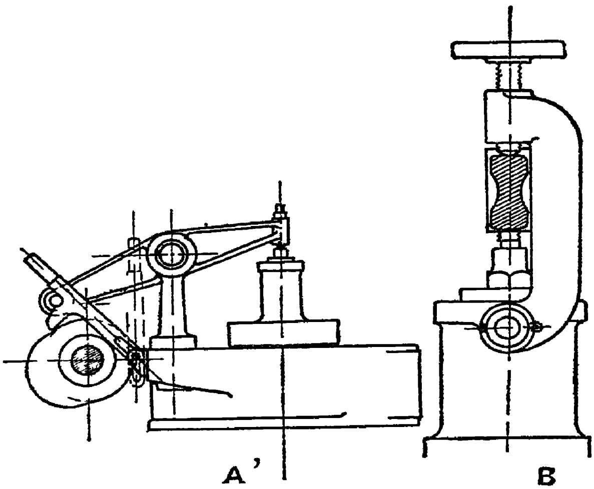
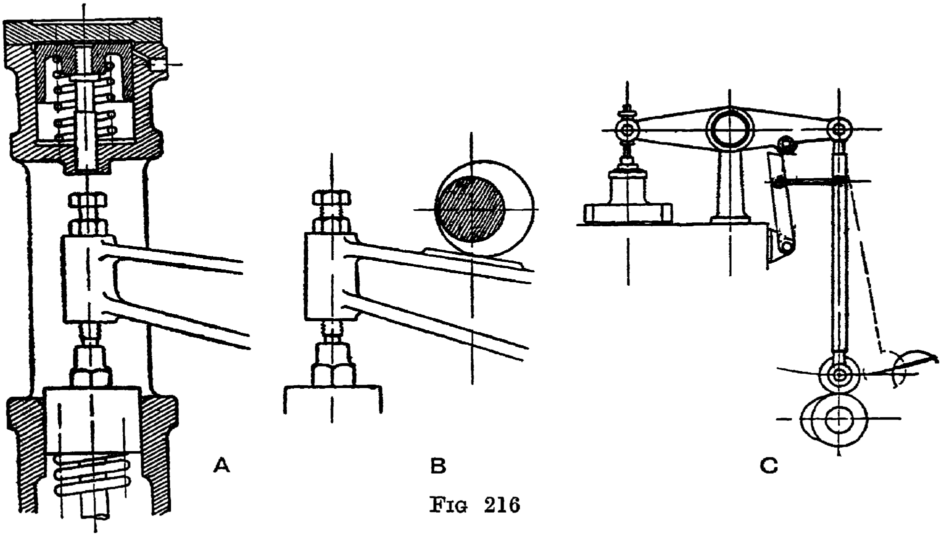


Fig 215

arranged above and in line with the valve Type B consists of a series of cams (one cam over each exhaust valve lever) mounted on a shaft running from end to end of the engine A turning movement of this shaft simultaneously depresses all the exhaust valves Type C is appropriate to those engines in which the valve levers are operated by push rods The first movement in reversing consists of swinging the lower end of the push rods out of the range of operation of the cams By a scheme of linkwork which is obvious from the illustration the same movement introduces a lever with an inclined face under a roller provided for this purpose attached to the valve lever with the result that the valve is held off its seat until the push rods regain their normal working positions

Proportions of Exhaust Valves and Casings —The diameter of the exhaust valve is almost invariably made the same as

that of the aır valve ın oıdeı to simplıfy manufactuıe and gıve a symmetrıcal arrangement of valve gear On the exhaust stıoke the area thıough the valve ıs more than suffıcıent but a faırly early openıng (40 to 50 before bottom dead centre) ıs necessary to effect a rapıd fall of pressure The dımensıons of the varıous parts of the valve and casıng are wıth a few exceptıons matters of experıence only and the approxımate proportıons gıven below wıth ıefeıence to Fıg 217 are representatıve of average pıactıce The figuıes are ex pressed ın terms of the valve dıameter as unıt



Fɪɢ 216

The larger figures refer generally to the smaller sızes of valve The sprıng whıch wıll be consıdered later has usually between twelve and twenty two turns A large numbeı of turns reduces the range of varıatıon of stress and consequently ıncıeases resıstance to fatıgue

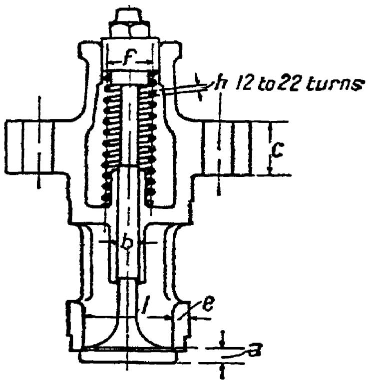
On the other hand a sprıng wıth a small number of turns has less tendency to buckle

The seatıng of the casıng ın the cylındeı cover ıs almost always made square now ınstead of conıcal as formerly The wıdth of seatıng need not exceed a quarter of an ınch ın the largest sızes The seatıng of the valve ın the casıng ıs usually made at an angle of 45 or 30 and the wıdth of the seatıng varıes greatly ın dıfferent desıgns Narrow seatıngs about an eıghth of an ınch ın wıdth appear to be least subject to pıttıng and seatıngs as narrow as 1/25th of an ınch have

been used successfully For large marine engines seats about $\frac{3}{8}$ inch wide seem to be preferred The spindle clearance may be about 20/1000 and the guide piston clearance about 10/1000 in all sizes

The size of the holding down studs may be found from the standard table (page 129) the total load being based on a pressure of 500 lb per sq in over the least area of the casing where it makes joint with the cylinder cover

The casing lugs should be amply proportioned particularly in cases where the castings are not above average quality and they should have a good hold on the cylindrical part of the casing reinforced if necessary by internal ribbing It is not



a	b	c	e	f	h
0 15 to 0 18	0 20 to 0 24	0 4 to 0 56	0 07 to 0 17	0 46 to 0 64	0 06 to 0 08

Fig 217

unknown for these lugs to break off in tightening up the studs so it is as well to err on the strong side

Exhaust Valve Springs —Apart from the weight of the valve in itself there are causes tending to open the exhaust valve when it should be shut viz —

(1) The vacuum on the suction stroke
(2) With four cylinder engines especially the first rush of exhaust from the neighbouring cylinder if the latter discharges into the same collecting pipe

In addition the inertia of the valve and any levers rods etc in connection therewith tend to make the latter lose contact with the operating cam in the neighbourhood of maximum lift These various influences are overcome by fitting a spring the normal load of which is equivalent to a

pressure of about 10 lb per sq in in valve area in the case of slow speed engines and anything up to about 20 lb per sq in or more in the case of high speed engines A method of computing the inertia effect will be dealt with in some detail as there is a tendency for higher speeds to be used in practice and the principles involved have a wide application in the design of high speed machinery generally

Inertia Effect of Valves —Consider the simple arrangement shewn in Fig 218 consisting of a valve directly operated by a cam without intermediate members Let the weight of the valve guide and roller etc be W lb The effect of the inertia of the spring may be allowed for by adding one third of its weight to that of the other parts Let x be the distance in inches of the valve from its seat at any instant If the shape of the cam and the speed of the cam shaft be known it is possible to express x in terms of the time t in seconds counted from the instant at which the valve begins to lift by means either of an equation or a graph exhibiting the lift on a time base If this equation (equation of motion) is available then one differentiation with respect to t gives an expression for the velocity denoted by $\dot{x}$ and a second differentiation gives the acceleration denoted by $\ddot{x}$ If the relation between x and t is given by means of a graph then the differentiation may be done by one or other of the graphical methods explained in books on practical mathematics In either case x being measured from the valve seat outwards positive values of $\ddot{x}$ denote inertia effects tending to press the roller against the cam and negative values of $\ddot{x}$ denote inertia effects tending to cause the roller to lose contact with the cam Here we are only concerned with the negative values of $\ddot{x}$ If X denotes the resultant force on the valve neglecting all effects except those due to the inertia

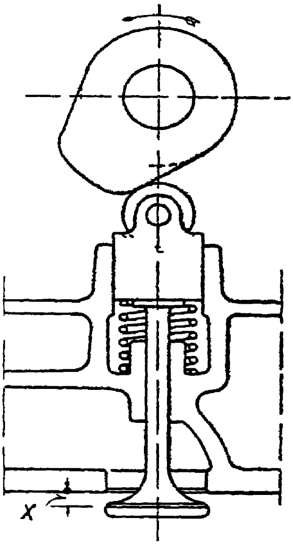


FIG 218

Then $$X = \frac{W}{g}\ddot{x}_{(\text{MAX})} \qquad (1)$$

$$g = 386 \text{ in /sec}^2$$

Equation (1) gives the minimum value of the spring pressure to prevent the roller jumping due to inertia

The value of x (max) is very easily calculated in one case viz when the cam is so designed that the valve describes simple harmonic motion that is when the graph of x and t is a sine curve In this case it is convenient to measure x from the position of mid lift positive outwards and negative inwards The equation of motion is then —

$$x = A \sin pt \qquad (2)$$

where A = Half the maximum lift in inches

and p is a constant such that $pT = 2\pi$

where T is the whole period of opening in seconds If the exhaust valve is open for 240 crank shaft degrees then —

$$T = \frac{60}{n} \times \frac{240}{360} \text{ and } p = \frac{2\pi\, n \times 360}{60 \times 240} = 0\ 157n \qquad (3)$$

n being the number of revolutions of the engine per minute

From (2) $x = -Ap^2 \sin pt$

And $x_{(MAX)} = -Ap^2$

Substituting in (1) $X = -\frac{W}{g} A p \qquad (4)$

Example W = 10 lb

A = Half lift = 0 375

n = 400 R P M

From (3) $p = 0\ 157 \times 400 = 62\ 9$

From (4) $X = \frac{10}{386} \times 0\ 375 \times 62\ 9^2 = 38\ 5$ lb

The valve would therefore require to be provided with a spring capable of exerting a force of 48 5 lb to deal with inertia and dead weight only apart altogether from gas pressure and friction

The use of the harmonic cam to which the above figures apply has not become very general in Diesel Engine practice the tangent cam shewn in Fig 249 Chapter XIV being more commonly employed Typical velocity and acceleration curves for a tangent cam are shewn in Fig 219

Effect of Levers and Push Rods etc —The simple case considered above of a cam operating directly on the end of the valve is seldom realised in practice and a somewhat more general case illustrated diagrammatically in Fig 220 will be considered We have now several different members all participating in the motion and acquiring momentum which

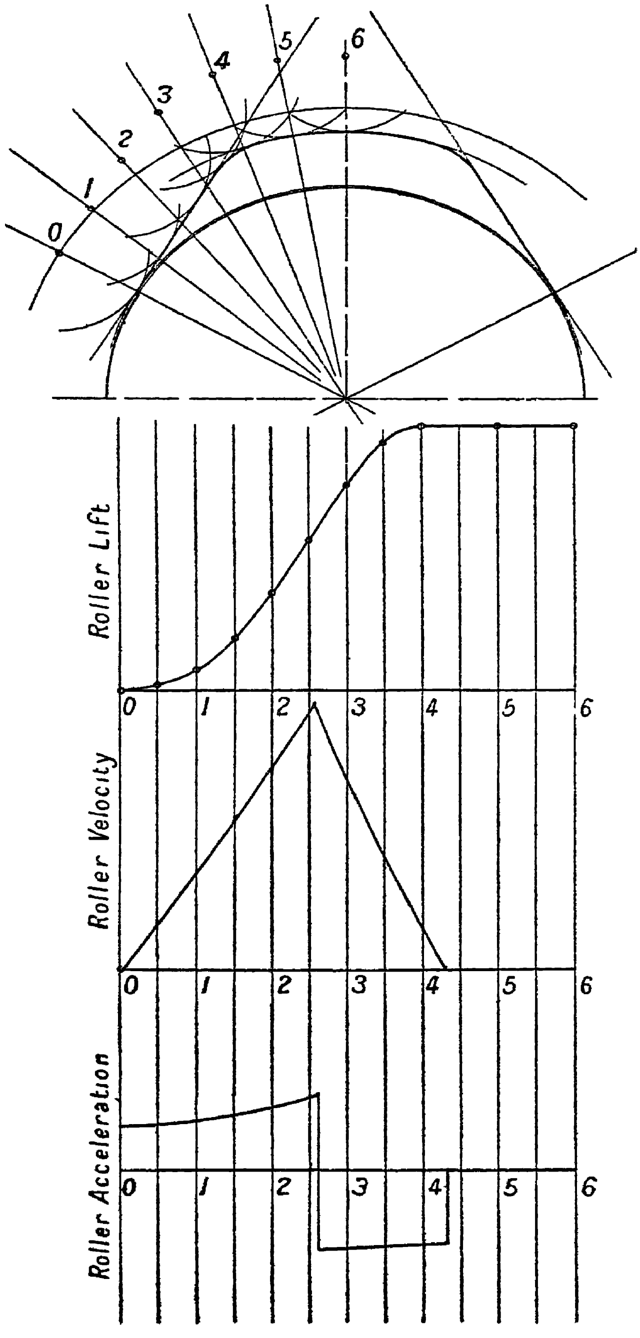

0
1
2
3
4
5
6
Roller Lift
Roller Velocity
Roller Acceleration
0 1 2 3 4 5 6
0 1 2 3 4 5 6
0 1 2 3 4 5 6

FIG 219

must be overcome by the spring The problem is a simple case of the general theory of a system of one degree of freedom simplified by treating the coefficient of inertia as a constant instead of a function of x Consider any particle of the lever AB situate at a distance r from the fulcrum E and having

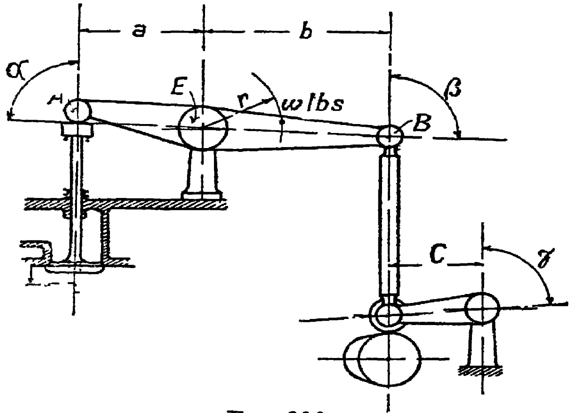


Fig 220

a weight w lb If s denote the speed of this particle during any small displacement of the system then —

$$s = x\frac{r}{a} \text{ and } s = x\frac{r}{a}$$

The effective force acting on the particle is therefore equal to $\frac{w}{g}x\frac{r}{a}$ and the reaction X_1 at A due to all such particles of the lever is given by

$$X_1 = \frac{x}{g\,a}\Sigma w\,r = \frac{x}{g} W_1 \frac{k_1}{a} \qquad (5)$$

Where W_1 is the weight of the lever and k_1 is its radius of gyration about E The expression $\frac{W_1 k_1^2}{g\,a^2}$ is the inertia co efficient of the lever with respect to the co ordinate x and may be denoted by A_1 The total reaction X at A due to the inertia effects of the valve itself all the levers push rods etc is the sum of all the reactions due to the individual members and therefore

$$X = x\,(A_0 + A_1 + A \;+ A_3) \qquad (6)$$

$$A_0 \text{ being} = \frac{W_0}{g} \text{ where } W_0 = \text{weight of valve}$$

and A_2 is the inertia coefficient of the push rod and A_3 is that of the link

By similar reasoning to that given above for A_1 it is found that $A_2 = \frac{W_2}{g}\left(\frac{b}{a}\right)^2$ where W_2 = weight of push rod

and $A_3 = \frac{W_3}{g}\left(\frac{k_3 b}{ac}\right)^2$ where W_3 = weight of link and k_3 = its radius of gyration about its axis

It will be seen at once that equation (6) is similar to (1) with inertia coefficient substituted for mass The assumption made is that the angles α β γ do not deviate far from 90 degrees For ordinary practical purposes a deviation of 10 or 15 degrees on either side involves a negligible error

Example	$W_0 = 6$ lb	$a = 10$	$k_1 = 7$
	$W_1 = 15$ lb	$b = 12$	$k_3 = 5$
	$W_2 = 8$ lb	$c = 7$	
	$W_3 = 5$ lb	$x = 1600$ in /sec ²	

$$X = x(A_0 + A_1 + A_2 + A_3)$$
$$= \frac{1600}{386}\left[6 + 15\left(\frac{7}{10}\right)^2 + 8\left(\frac{12}{10}\right)^2 + \left(\frac{12 \times 5}{10 \times 7}\right)\right]$$
$$= 118 \text{ lb}$$

For slow running engines the inertia does not usually amount to more than two or three lb per sq in of valve area In most cases the spring may be based on the inertia load plus about 6 lb per sq in of valve area to deal with the other effects which enter into the question

Strength and Deflection of Springs —The usual formulæ for the safe load and the deflection of springs made of steel wire of circular section are given below for handy reference

$$P = 0{\cdot}2f\,\frac{d^3}{r} \quad \text{Where} \begin{cases} P = \text{Safe load (maximum) in lb} \\ f = \text{Safe stress usually about 60 000 lb per sq in} \\ d = \text{Diameter of wire in inches} \\ r = \text{Mean radius of coils in inches} \end{cases}$$

And

$$\delta = \frac{64\,n\,r^3}{d^4} \times \frac{P}{G} \quad \text{Where} \begin{cases} \delta = \text{Deflection in inches} \\ n = \text{Number of turns (free)} \\ G = \text{Modulus of rigidity usually taken to be about 12 000 000 lb per sq in} \end{cases}$$

Exhaust Piping —A common arrangement of unjacketed cast iron exhaust piping is shewn in Fig 221 The flexibility of the system renders any special provision for expansion unnecessary The piping itself being out of reach need not be lagged and may be as light as casting considerations will allow

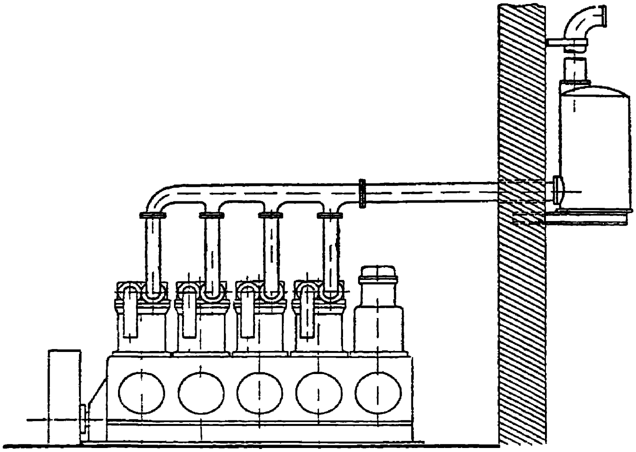


Fig 221

The connecting pieces between the various covers and the common discharge pipe may be made equal in bore to the diameter of the exhaust valve The bore of the collector pipe joining the silencer may be proportioned with reference to the nominal velocity of the exhaust gases as follows —

Let V_p =Piston speed in feet per second
V =Nominal speed of exhaust in feet per second
B =Bore of cylinder in inches
d =Bore of exhaust pipe in inches
n =Number of cylinders

Then
$$V = V_p\left(\frac{B}{d}\right)^2 \times \frac{n}{4}$$

In using this formula n should be put equal to 4 in all cases where the number of cylinders is equal to or less than 4 The reason for this is that the gases discharge intermittently and an engine of one two or three cylinders requires approximately the same size of pipe as a four cylinder engine of the same size and speed The value of V_x varies from about 70 in

small engines to 110 in large The pipe leading from the silencer to the atmosphere may be made about 25% larger in the bore A somewhat neater arrangement involving an inter mediate collector under the floor is shewn in Fig 222 The individual exhaust pipes must now be water cooled but there is no objection to short uncooled sections in way of the flanges of sufficient length to accommodate the bolts If the pipes are cast the thickness of the outer walls need not exceed about $\frac{1}{4}$ to $\frac{3}{8}$ with good foundry work The jackets may also be of welded steel tubes or sheets

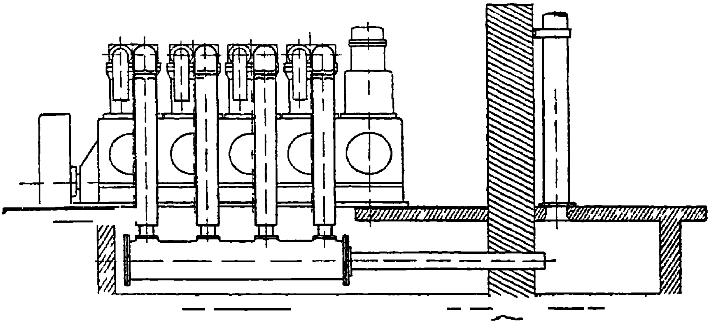


Fig 222

In marine installations the exhaust is sometimes used to furnish a supply of hot water for heating and other purposes and this may be achieved by providing the exhaust collector with nests or coils of tubes through which water is circulated As a rule the exhaust from a four stroke marine engine is not sufficiently hot to necessitate the provision of a water cooled silencer apart from the arrangements which have just been mentioned

Silencers —As a rule four stroke Land Diesel Engines are supplied with a cast iron silencer having a capacity of about six times the volume of one of the working cylinders and a common type is shewn in Fig 223

The result is not always satisfactory and better results are obtained by using a large underground brick or concrete chamber having twenty or thirty times the volume of one cylinder Wrought iron silencer unless water jacketed or buried underground usually give out a ringing noise unless the gases on entry are made to diffuse through a trumpet arrangement similar to that described above under suction pipes

Two Stroke Engines Scavengers —If the efficiency of the scavenging process could be definitely ascertained in every case it would be a simple matter to calculate a suitable capacity for the scavenge pump In those engines of which the results on trial suggest that this process is nearly perfect the scavenge pump appears to have a stroke volume capacity of about 1 4 times the aggregate stroke volume capacities of the cylinders which it feeds Some of the earlier designs were based on ratios as high as 1 8 or 2 Such ratios penalise the design in two ways firstly by unduly increasing the size of the scavenger and secondly by raising the pressure required to pass the scavenge charge through the ports Good mechanical efficiencies can only be secured by reducing the scavenge work to a minimum In the best modern designs low pressures of 2 lb /in ² or less are usual In order to predict the scavenge pressure in advance some such calculation as that indicated in Chapter III is required unless direct experience is available

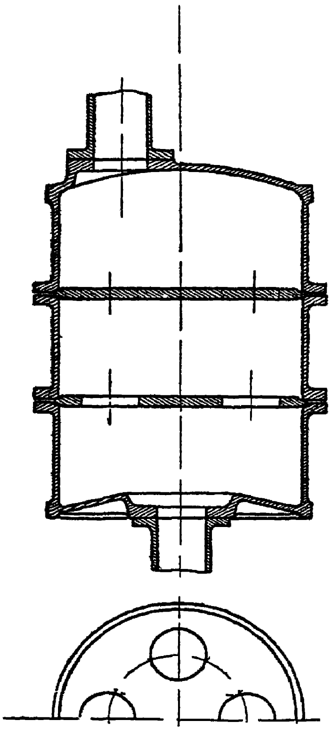


Fig 223

The work done by the scavenger is all lost work and must therefore be kept at a low figure On the other hand the super pressure above that of the atmosphere with which a controlled scavenge engine starts the compression stroke is a valuable factor in increasing the power of a given sized cylinder If simplicity is the main consideration and uncontrolled port scavenge is adopted it becomes necessary to be content with a very moderate M I P as the super pressure obtainable at the beginning of compression appears to be very limited with the arrangements hitherto adopted on such engines

Construction of Scavengers —It is not necessary to deal exhaustively with the details of scavenge pumps as these differ little from the corresponding parts of L P steam engines The scavenger or scavengers are preferably driven off cranks provided for the purpose on the main shaft as in Fig 224 In

marine designs the link drive shewn in Fig 225 has also been used but as usually carried out is open to the following objections —

(1) The arrangement does not lend itself readily to the closed engine type of framework which appears to have every advantage for Diesel Engine work

(2) The side levers involve cantilever connections which lack rigidity

(3) The heavy reversals of thrust are liable to cause knocking and vibration owing to small bearing surfaces and poor lubrication

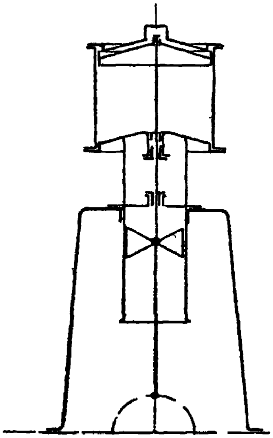


Fig 224

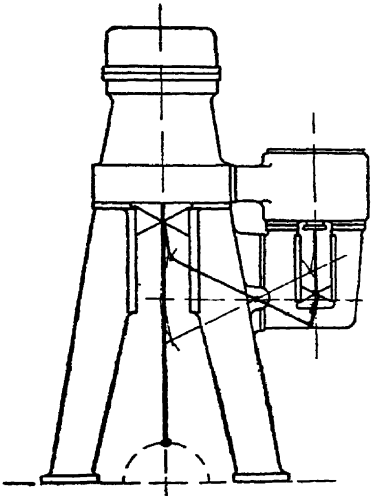


Fig 225

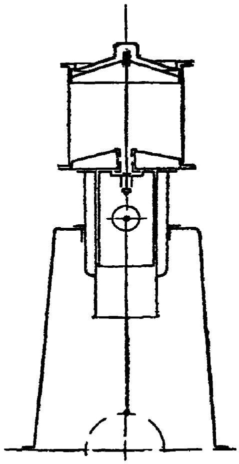


Fig 226

A scheme suitable for land work but a trifle inaccessible for marine purposes consists of a tandem arrangement of scavenger and low pressure stage of blast air compressor as in Fig 226

Valve Gear —The earlier Diesel scavengers were fitted with piston valves single or double ported and this type of gear is still retained in some designs For reversing such valves the Stevenson link motion appears to be the best solution on hand at present In other designs automatic disc or plate valves are being increasingly used with a gain in simplicity and the advantage in marine work that no reversing gear is necessary The liability of the valves to failure by fatigue seems to be their chief drawback

Scavenge Air Receivers —From the scavenge pump the air

passes to a receiver in communication with the cylinder covers or scavenge air belts of the working cylinders With the usual arrangement where one or at most two double acting scavenge pumps are used to supply a number of working cylinders say three to eight it is advisable to make the capacity of the receiver large compared with the stroke volume of one scavenger in order that the pressure in the receiver may remain sensibly constant and a suitable capacity may usually be secured by making the diameter of the receiver about 1 to 1 3 times the cylinder bore In any proposed case it is a simple matter to construct a diagram shewing on a time base the rate at which air is being passed to the receiver by the scavenger and carried away from it by the working cylinders The resultant effect in creating fluctuations in a receiver of any proposed capacity is then easily calculated The effect of too small a receiver capacity is to give those cylinders which begin compression at the instant of maximum scavenge pressure an advantage over those less favourably timed The effect is readily studied in practice by means of light spring diagrams taken from the receiver and the working cylinders respectively By way of example if two cylinders start compression with absolute pressures of 21 and 20 lb per sq in respectively then the first will (other things being equal) have a maximum power capacity 5% greater than the second or if they are worked at the same power the second cylinder will work with a mean absolute charge temperature 5% greater than the first which is no small evil In some two stroke designs one double acting scavenger is provided for each pair of working cylinders and if the deliveries are correctly timed with respect to the scavenge periods the receiver capacity does not require to be very large Scavenge receivers are usually made of riveted or welded sheet steel having a thickness of about one per cent of the diameter A disastrous explosion at Nuremberg in 1912 traceable to the ignition of lubricating oil vapour in the scavenge receiver of a large two stroke engine points to the desirability of providing drain cocks and a relief valve or safety diaphragm of large area The area of the trunk communicating with each cylinder should be well in excess (usually about double) of the maximum aggregate area of the valves or ports which it supplies

Scavenge Valves—In different designs embodying the principle of scavenging through the cover one two three and four valves have been employed In particular if two valves

are used they may be identical with the air and exhaust valves used in four stroke engines of the same size with the piston speeds at present customary With any other numbei of valves the area may be made equivalent In view of the

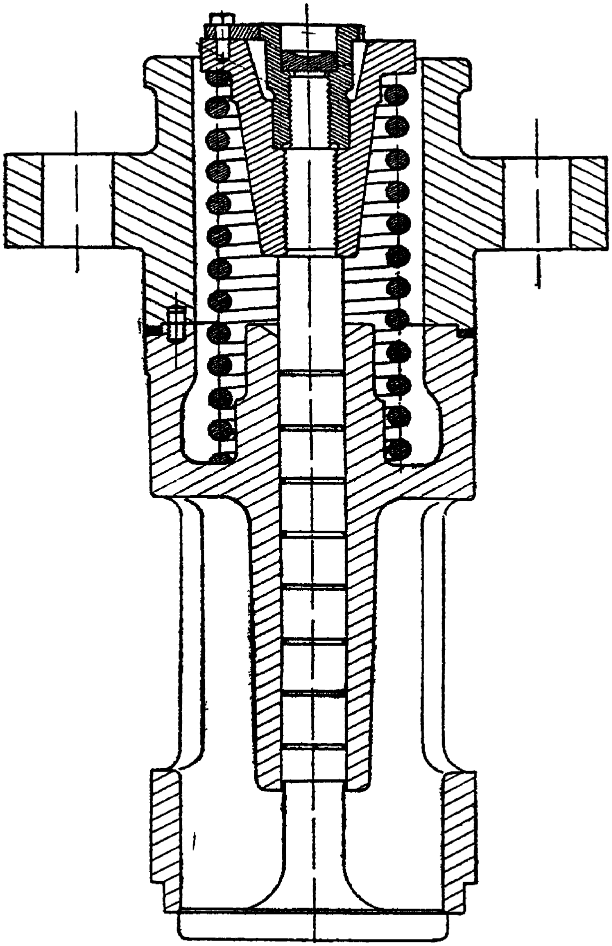


Fig 227

relatively light duty which devolves upon them scavenge valves are usually made of somewhat simpler construction than exhaust valves as shewn in Fig 227 To prevent loss of scavenge air past the spindle the latter is usually made a good fit

With controlled port scavenge double beat valves piston

valves and Corliss type valves have all been used in different designs as shewn diagrammatically in Fig 228 The use of the valve is exclusively to regulate the instant at which air is admitted to the cylinder the point of cut off being determined by the piston covering the ports on the up stroke

It therefore follows that so long as the valve is full open at the instant when the ports are covered the point at which the valve seats again may be determined arbitrarily by other considerations affecting the valve gear

Ezhaust System —The chief evil to be guarded against in the design of the exhaust system of a two stroke engine is the interference of the exhaust rush of one cylinder with the scavenge process of another An obvious but inconvenient way of avoiding this trouble is to provide separate exhaust

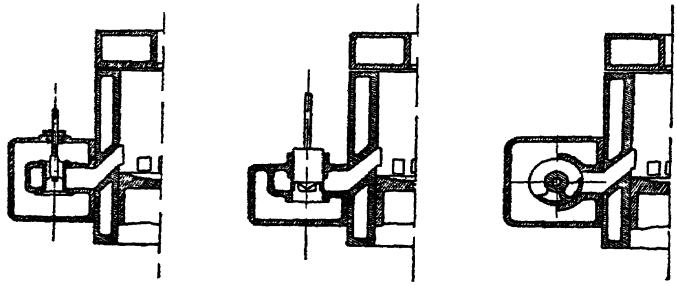


FIG 228

pipes and silencers for each cylinder Practically the same effect can be achieved by providing common exhaust systems for pairs of cylinders whose cranks are at 180° This however does not appear to be necessary and satisfactory results with an exhaust system common to all cylinders are obtained by the use of an arrangement similar to that shewn in Fig 222 for four stroke engines

The essential point is that the pipes which connect each cylinder to the common collector should be of large diameter (about three quarters of the cylinder bore) and that the collector itself should have a volume large in comparison with the stroke volume of one cylinder It does not follow however that a free passage for the exhaust is a necessary condition for efficiency as the latter has sometimes been improved by the insertion of a throttling diaphragm in the exhaust passage at the point where the pipe joins the cylinder

Silencers —The sudden release by the uncovering of ports of a pressure of 40 lb per sq in and upwards produces a noise which in the absence of a silencer can be heard some miles off A type of silencer which renders the exhaust inaudible a few yards away without imposing any back pressure is shewn in Fig 229 This type of silencer is most effective when water

Diameter of Silencer	3 5 B
Length	5 B
Bore of Inlet Pipe	0 55 B

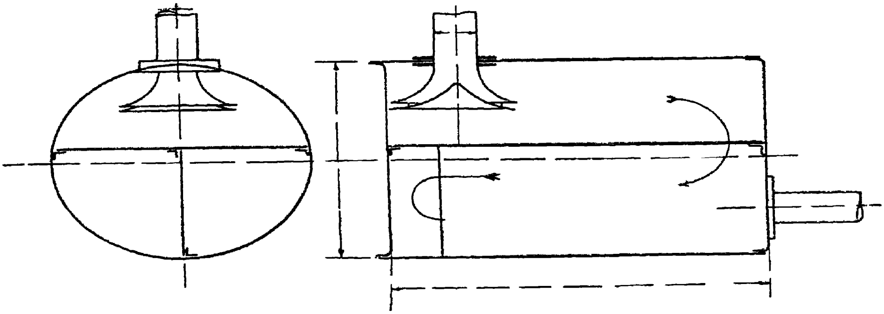


Fig 229

jacketed Approximate main dimensions are given in terms of the bore of the engine cylinder on the assumption of a piston speed of about 800 feet per minute For land purposes a large pit without special baffles would probably serve equally well

Literature —For information on the mechanics of cam operated mechanism see —

Goodman J Mechanics Applied to Engineering (Longmans)

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Also numerous articles in the *Automobile Engineer*

CHAPTER XIII

COMPPESSED AIR SYSTEM

As mentioned in Chapter I the injection of fuel by means of an air blast is one of the outstanding characteristics of the Diesel Engine and it seems probable that its use in the early experimental engines was suggested by the compressed air apparatus used to start the engine and which still appears to be the most practicable method of doing this The utility of the air blast is by no means confined to its function of injecting the fuel in fact the widespread use of mechanical means of injection in other types of oil engine clearly indicates that effective atomisation can be obtained otherwise As Guldner has pointed out the use of an air blast probably secures a more efficient mixing of the cylindei contents than could be obtained in any other practicable manner The advantages in efficiency which such mixing secures are easily appreciated on examination of the thermodynamic principles involved With good mixing the combustion proceeds rapidly and reduces after burning to a minimum and fuither the whole charge tends to remain homogeneous as to temperature a necessary condition for maximum efficiency In practice there are two aspects from which the efficient utilisation of heat should be viewed viz —

(1) Economy in fuel consumption
(2) The effect of efficient combustion in keeping the mean cycle temperature to a minimum

The last consideration is a vital one from the point of view of reliability and durability and experience abundantly proves that any considerable increase of the cycle temperature due to overloading leakage past valves loss of volumetric efficiency or other causes is sufficient to convert an otherwise reliable machine into a source of trouble

The blast air has a pressure which varies from about 900 to 1000 lb /sq in at full load to about 600 lb /sq in at no load and in land engines is usually supplied by a compressor forming an integral part of the engine In marine installations the compressors are sometimes driven by separate auxiliary engines An arrangement adopted by one maker was to drive the lower stages of the compressors by auxiliary engines the last stage being performed by a high pressure plunger driven by the main engine From the compressor the air passes via coolers to a blast reservoir or bottle of sufficient capacity to absorb fluctuations of pressure and fitted with suitable distributing valves one of which communicates with the fuel injection valves and another enables surplus air to be passed to the storage reservoirs provided for starting purposes Cam operated valves in the covers of one or more cylinders enable the stored air to be used to give the engine the initial impetus which is necessary before firing can begin With land engines the starting bottles are generally charged to a pressure of about 900 lb per sq in and with the fly wheels commonly used it is not necessary to provide starting valves for more than one cylinder out of three With marine engines storage pressures of about 350 lb per sq in are more common on account of the difficulty of making high pressure reservoirs of large size and in order to secure prompt starting from any position starting valves are fitted to every cylinder

The air system also includes certain servo motors or air engines frequently used to perform operations of reversing the valve gear The various organs will now be considered in more detail

Air Compressors —Four stroke land engines of the slow speed type as at present constructed require compressors having a capacity of about 15 cubic feet per B H P per hour which assuming a volumetric efficiency of 80 % corresponds to a stroke volume capacity of about 19 cubic feet per hour Small high speed engines appear to require about 25 % more than this allowance The above method of basing the compressor capacity on the B H P is not a very satisfactory one as different makers have different views as to power rating A better plan is to express the L P stroke volume as a percentage of the aggregate cylinder volume and the following figures are representative of average practice for land engines In view of the demands made on the system when manœuvring

marine installations are provided with a special manœuvring air compressor separately driven

Bore of working cylinders	Ratio L P stroke vol —Stroke vol of working cylinders	
	Four Stroke Engines	Two Stroke Engines
10	0 08	0 16
15	0 07	0 14
20	0 05	0 09

Number of Stages —For small slow running compressors two stages are sufficient but a 9 inch diameter of low pressure cylinder appears to be about the safe limit and even with this restriction it appears wise to abandon the principle of equal distribution of work between the stages The small diameter of the H P cylinder affords little cooling surface for the dissipation of heat and this consideration points to the advisability of arranging for the greater part of the work to be done in the L P stage a conclusion which has been anticipated by experience

Three stage compressors are being increasingly used even for the smaller sizes but four stages have not come into general use even in the largest sizes With three or four stages the principle of equal division of work is open to less objection owing to the smaller ratio of compression in each stage

Compressor Drives —Almost every conceivable type of drive has been adopted at one time or another and only the commonest are mentioned below —

(1) Tandem two or three stage compressor driven off the crank shaft This arrangement appears to have the balance of advantages for most purposes

(2) Tandem two stage compressor driven by links and levers from each connecting rod or crosshead This arrangement is expensive but has the advantage of distributing the work amongst a number of small compressors which are subject to less heat trouble than one compressor of the same capacity The suction pressure of the H P stage is usually sufficient to prevent reversal of thrust due to inertia and consequently sweet running is secured

(3) Sımılar to (2) but stages separate Thıs arrangement ıs bad as the coolıng surface ıs less than ın case (2) and the L P gear ıs subȷect to reveısal of thrust due to ınertıa

(4) Twın tandem three stage compressors drıven off a paır of cranks at the forward end of the engıne Thıs ıs a very suıtable arrangement for large marıne engınes

Constructıve Detaıls —The constructıon of aır compressors beıng a specıalısed branch of mechanıcal engıneerıng ıt ıs not proposed to gıve here more than a very brıef reference to the subȷect The cylınders of tandem two and three stage machınes are frequently cast ın one pıece ıncludıng the water ȷacket The relatıvely low temperatures obtaınıng ȷustıfy thıs pro cedure provıded sound castıngs can be obtaıned wıth reason able regularıty The foundry work may be sımplıfied ın the case of two stage compressors by the followıng dıvısıon of materıal —

L P Cylınder and Jacket—one castıng
L P Cover and H P Jacket—one castıng
H P Lıner and H P Cover—separate castıngs

One advantage of thıs scheme ıs the possıbılıty of renewıng the H P lıner when worn The latteı ıs pecuharly subȷect to rapıd wear on account of the hıgh pressure behınd the ıings

Sımılar arrangements are of course possıble wıth three stage machınes The possıbılıty of ıncreasıng the coolıng surface of the H P lıner by means of rıbs does not appear to have receıved very much attentıon The trunk pıstons serve as admırable crossheads beıng almost entırely free from the heat trouble to whıch the pıstons of ınternal combustıon engınes aıe subȷect For the L P and ınter medıate stages Ramsbottom rıngs are usually fitted In very large machınes the latter can also be fitted to the H P plunger Small H P plungers are usually fitted wıth some arrange ment sımılar to that shewn ın Fıg 230 The only thıng whıch need be saıd about the connectıng rods ıs that on account of the thrust beıng always ın one dırectıon specıal care ıs requıred ın the detaıls of lubrıcatıon

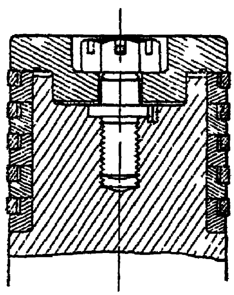

FIG 230

The design of valves has an important bearing on the success or failure of a compressor The chief evils to be avoided are —

(1) Sticking of the valves off their seats due to deposits of carbonised oil

(2) Damage to valves or valve seats due to hammering

The first is influenced more by the efficiency of the cooling arrangements and the compression ratio than with the design of the valves themselves For obvious reasons the H P valves are most subject to this trouble

The second trouble is usually due to the valves being too heavy having too much lift or the failure to provide adequate cushioning and in successful designs is avoided by one or more of the following means —

(1) Making the valves in the form of very light plates or discs with a very small lift

(2) Providing a large number of very small valves in place of one or two large ones

(3) Where large valves of considerable weight are used arranging for some sort of dash pot action

All the valves should be easy of access and removal Experiments with existing types of compressor seem to indicate that makers are inclined to base their valve dimensions on an air speed very much lower than is necessary One or two per cent loss of efficiency is of small importance if such a sacrifice enables the size of the valves to be reduced

In some designs the intercoolers are separate from the compressor cylinder and in others take the form of pipe coils arranged round the compressor cylinders inside a removable water jacket Vibration of the coils should be prevented by adequate clamps and stays and no sharp bends are allowable on account of a scouring action (presumably due to turbulent flow) which in acute cases may cause fracture of the pipe in a short time L P intercoolers are sometimes made similar to tubular condensers and in other designs take the form of a cast iron vessel provided with internal helical baffles which give rise to turbulent flow and increase greatly the efficiency of the cooling surface It is desirable in all cases to fit a final cooler to reduce the temperature of the fully compressed air before entering the blast receiver Each receiver should be fitted with safety valve and drain One or two isolated cases of explosion traceable to accumulation of lubricating oil in the

intercooler system emphasise the necessity for these fittings Some makers fit special purge pots in communication with each receiver for the collection and discharge of condensed water and oil

Calculations for Compressors—In calculating the L P stroke volume required to furnish a given free air capacity allowance must be made for the volumetric efficiency which depends mainly on the clearance space and the delivery pressure For example suppose the clearance to be 3% of the stroke volume and the receiver pressure to be 150 lb per sq in On the suction stroke the suction valve will not begin to lift until the air left in the clearance space has expanded down to atmospheric pressure If this clearance air expands according to the law —

$$P V^{1\ 2} = \text{constant}$$

then its expanded volume expressed as a percentage of the stroke volume will be —

$$3 \times \left(\frac{164\ 7}{14\ 7}\right)^{\frac{1}{1\ 2}} = 22\ 5\%$$

Subtracting its original volume viz 3% the amount by which the effective stroke is shortened is 19 5% and the volumetric efficiency is 80 5% A further deduction should strictly be made for the fact that at the beginning of the compression stroke the cylinder contents are in general at a pressure slightly less than atmospheric One or two per cent will usually cover this contingency This example is sufficient to shew the importance of reducing the clearance volume of the L P cylinder to a minimum The efficiencies of the L P or H P cylinders may be found similarly but only influence the volumetric efficiency of the compressor as a whole indirectly by raising the receiver pressure above the value it would have if there were no clearance It will be evident on reflection that leakage past the H P delivery valves will also raise the receiver pressures and for this reason it is desirable to fit pressure gauges to all receivers so that the condition of the valves may be inferred from the gauge readings

Assuming perfect intercooling and equal volumetric efficiency in all the stages the pressure of the atmosphere and the receiver pressures (absolute) will be in inverse ratio to the stroke volumes of the cylinders and equal division of work between the stages will be secured by the proportions given below —

	Atmospheric Pressure	Intermediate Pressure	High Pressure
Two Stages	1 at =14 7 lb /in	8 at =118 lb /in	64 at =940 lb /in

L P volume =8 H P volume

	Atmospheric Pressure	1st Intermediate Pressure	2nd Intermediate Pressure	High Pressure
Three Stages	1 at =14 7 lb / in	4 at =58 8 lb / in ²	16 at =235 lb / in	64 at =940 lb / in

L P volume I P volume H P volume =16 4 1

In practice better results are obtained with two stage machines by the following proportions —

Atmospheric Pressure	Intermediate Pressure	High Pressure
1 at =14 7 lb /in	12 at =177 lb /in	64 at =940 lb /in

L P volume =12 H P volume

Assuming that the L P stroke volume has been determined by some such considerations as the above the actual cylinder dimensions are found by selecting suitable values for the piston speed and the L P stroke to bore ratio The piston speeds commonly used lie between about 300 to 600 feet per minute the lower speeds being usually associated with small machines With a little care it is possible by making small variations in the strokes to design a series of four or five compressors suitable for engines covering a wide range of powers Such a scheme involves sacrifices in some cases which however would appear to be quite outweighed by the advantages of standardisation The ratio of stroke to bore of the L P cylinder will generally lie between about 0 7 and 1 7 The valve areas for each stage are based on some figure for the mean velocity obtained in the case of suction valve by multiplying the mean piston speed by the ratio of piston to valve area and in the case of delivery valves the mean piston speed during the delivery period by the same ratio Certain continental authorities recommend speeds not exceeding 80 and 115 feet per second for the suction and delivery respectively but it appears that these figures may be doubled or even trebled with impunity and sometimes to advantage

The calculations of the strength of the various parts are straightforward involving no special principles and are therefore passed over The cooling surface to be provided for intercooling is a very important matter and the following figures from a successful design may be useful as a basis of comparison in the absence of first hand experimental data

THREE STAGE COMPRESSOR Free air capacity 130 ft ^{3}/min

Cooling surface copper pipe coils	L P	8 7 ft ^{2}
	L P	3 6 ft ^{2}
	H P	3 6 ft ^{2}

The subject of heat transmission being a very important one in connection with internal combustion engines a brief reference to the usual theory is inserted below

Transmission of Heat through Plates —Referring to Fig 231 the direction of heat flow is indicated by the arrow and the symbols t_1 t_2 t_3 t_4 denote the temperatures of the hot fluid the hot side of the plate the cold side of the plate and the cold fluid respectively The total heat drop consists of three stages —

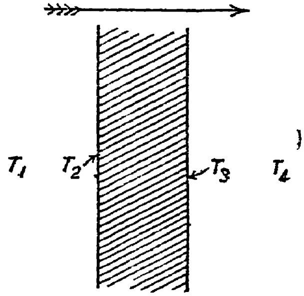

FIG 231

(1) An apparently sudden drop from the hot fluid to the plate
(2) A steady gradient across the thickness of the plate
(3) An apparently sudden drop from the plate to the cold fluid

The assumption is that the rate of heat flow is dependent only on the temperature drop the thickness of the plate and the particular fluids employed On this assumption if Q is the amount of heat transmitted per hour per unit of area then —

$$Q=\alpha_1(t_1-t_2)=\frac{\lambda}{d}(t_2-t_3)=\alpha_2(t_3-t_4) \qquad (1)$$

from which

$$Q=\frac{(t_1-t_4)}{\frac{1}{\alpha_1}+\frac{1}{\alpha_2}+\frac{d}{\lambda}} \qquad (2)$$

Where d = thickness of plate and α_1 α_2 and λ are constants

For compressed air in pipes α_1 is given approximately by —

α_1 = about 3 0 $\frac{w}{a}$ ($\frac{1}{2}$ pipe) to α_1 = about 2 2 $\frac{w}{a}$ (2 pipe)

where w/a = mass flow in lb per ft ^{2} of sectional area

For nominally still water α_2 = about 100 (B T U /hr ft ^{2} F)

Values of λ are given below for various metals —

Iron	460	B T U /ft ^{2} deg F per inch of thickness
Mild steel	320	
Copper	2100	
Brass	740	
Scale (average)	12	

The values of λ are well determined but unfortunately the term involving this constant is the least important of the three as the bulk of the heat drop occurs at the surfaces of the plates The values of a_1 and a_2 must be used with caution as the small amount of published data indicates that these constants are subject to enormous variation under different circumstances In particular the value of a_2 is greatly increased by the eddying motion produced by high speed flow through narrow channels or tubes If the water boils at the surface a may be about 2000 even if the water is nominally still

The value of a_1 for compressed air is somewhat higher at high temperatures than at low For further information on the subject of emissivities and heat transmission generally the reader is referred to the sources of information mentioned below [1]

Equation (2) deserves careful consideration as it contains the crux of most heat transmission problems The term involving d and λ is usually negligible (except in cases of very rapid transmission) unless due allowance is made for scale or deposit It sometimes happens that either a_1 or a_2 is almost negligible in comparison with the other

Air Reservoirs —The usual arrangement of air reservoirs for land engines is shewn in Fig 232 This scheme was devised in the very early days of the development of the Diesel Engine and no substantial improvement has been made on it in recent years Two starting and one blast air bottles are provided all designed for a working pressure of about 1000 lb per sq in One of the starting bottles serves as a reserve in case of a failure to start the engine due to any derangement In the

[1] High speed Internal Combustion Engines Judge Heat Transmission Report by Prof Dalby to the Inst Mech Engs 1909 Notes on Recent Researches paper by Prof Petavel Manchester Assoc of Engineers Oct 1915 The Laws of Heat Transmission Lecture by Prof Nicholson Junior Inst Jan 1909 Heat Transmission by Royds Constable

event of such failure every care is taken to make certain that the engine is in perfect order before using the reserve bottle and it seldom happens in practice that the bottles require to be replenished from outside sources of supply The connections between the bottles the air compressor and the engine should

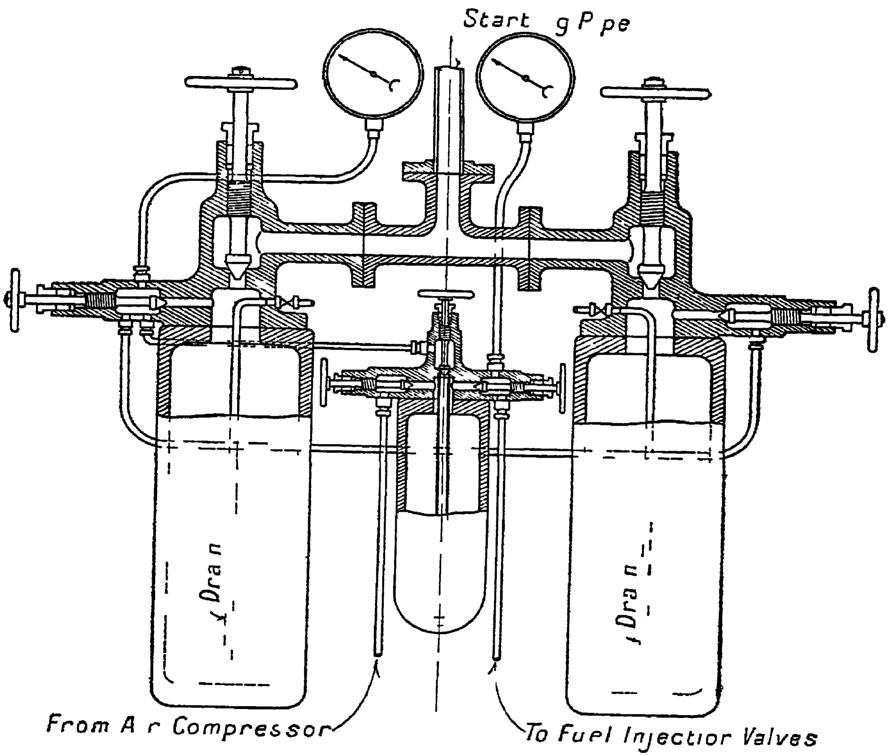

FIG 232

be quite clear from the diagram Only one or two points will be mentioned

(1) Before starting up it is possible to ascertain the pressure in each of the three bottles by opening up the appropriate valve on each bottle head in rotation In each case the pressure is recorded on the left hand gauge
(2) The pressure in any pair or all three bottles may be equalised by opening up a pair or all three such valves
(3) The right hand gauge registers the blast pressure on the engine side By throttling the blast control valve on

the blast bottle head the injection pressure may be regulated below that of the bottle This is done when replenishing the starting bottle on light load It is thus possible to pump up the starting vessel to 1000 lb /in ² whilst the blast pressure is only 600 lb /in ² as required for light running

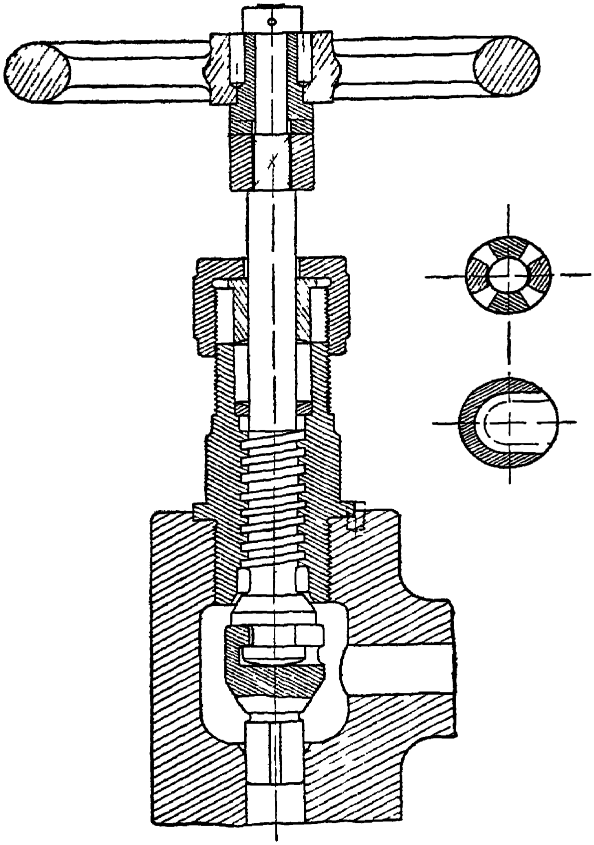

FIG 233

The bottle heads containing the various valves are usually machined from a solid block of steel A detail of one of the valves is shewn in Fig 233

The bottles themselves are of weldless steel and a neck is frequently screwed on as in Fig 234 Some idea of the capacities of the bottles commonly provided may be gathered from the following table —

TOTAL CAPACITY OF H P STARTING AIR BOTTLES
(four stroke engines)

Engines of about 9 *bore having one to six cylinders*—about fourteen times the stroke volume of one cylinder

Engines of about 24 *bore having one to six cylinders*—about seven times the stroke volume of one cylinder

Owing to the expensive machinery required to manufacture weldless reservoirs only a certain limited number of standard sizes are available at reasonable prices and in very large instal lations it is sometimes necessary to provide groups of four or more starting vessels The usual working stress is about

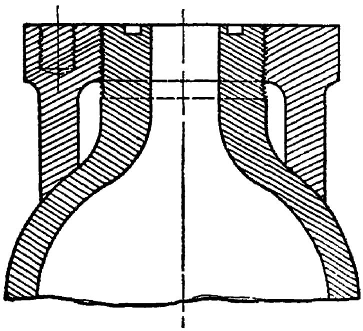

FIG 234

8000 lb /in ^{2} and it is customary to specify a water test pressure of double the working pressure

Riveted Air Reservoirs —For marine installations of high power it is usual to use a lower air pressure of about 350 lb /in ^{2} for starting purposes The air reservoirs now require to have a very much larger cubic capacity but the reduced pressure permits of the employment of riveted reservoirs The con struction of the latter need not be dealt with here being com parable with that of the steam drums of modern water tube boilers Adequate drainage for condensed water and oil vapour and also a manhole for inspection and cleaning should be provided These matters as well as others dealing with the strength of the riveted joints the quality of material to be used and the tests to be carried out on completion form the subject matter of regulations by the various insurance societies and the Board of Trade

Blast Piping System —From the blast bottle the injection

air passes to a main running along the back of the engine where it is distributed by short lengths of pipe to the several fuel valves as in Fig 235 Where one fuel pump is provided for a number of cylinders the fuel distributors may be made to serve as distributing tee pieces for the blast air In marine

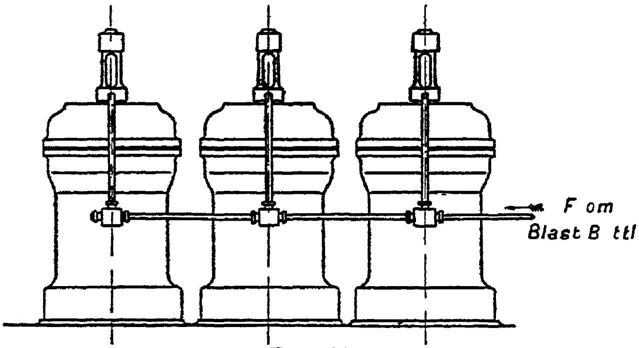

Fig 235

engines it is usual to provide a shut down valve as in Fig 236 to each tee piece so that the supply to any individual cylinder may be cut off to enable the fuel valves to be changed without stopping the engine

In addition it is sometimes necessary (see Chapter XIV) to provide a valve whereby the whole supply of blast air is automatically cut off when the manœuvring gear is put into the stop position The bore of the blast air supply pipe to each cylinder need not exceed about 2% of the cylinder bore but is usually greater than this in small engines to avoid the multiplication of standard sizes of unions The blast air main may be about 4% of the cylinder bore for any number of cylinders up to about six The same type of union may be used as has already been illustrated in Fig 170 in connection with the fuel system Other types of union are in use notably the Admiralty Cone Union which is also very serviceable

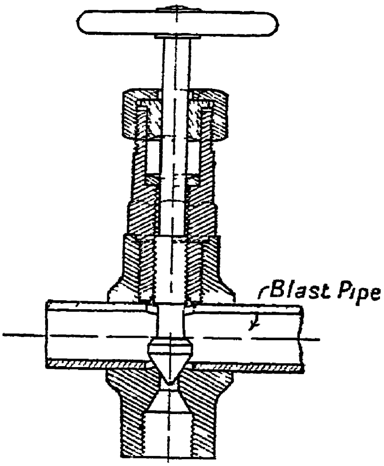

Fig 236

The Starting Air Pipe System —With land engines it is quite common to provide one cylinder only with a starting valve when the number of working cylinders does not exceed

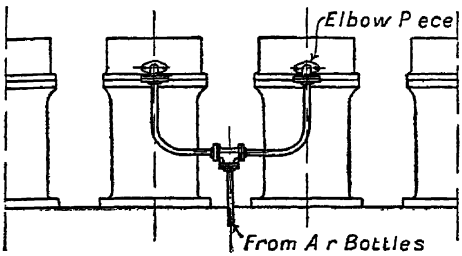

FIG 237

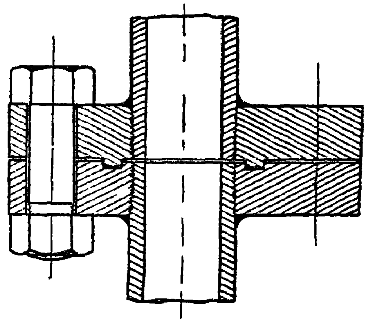

FIG 238

four With six cylinders and upwards two and sometimes three units are provided with air starting arrangements A neat arrangement of the starting pipe is shewn in Fig 237 for a four cylinder engine In this case the design of the starting

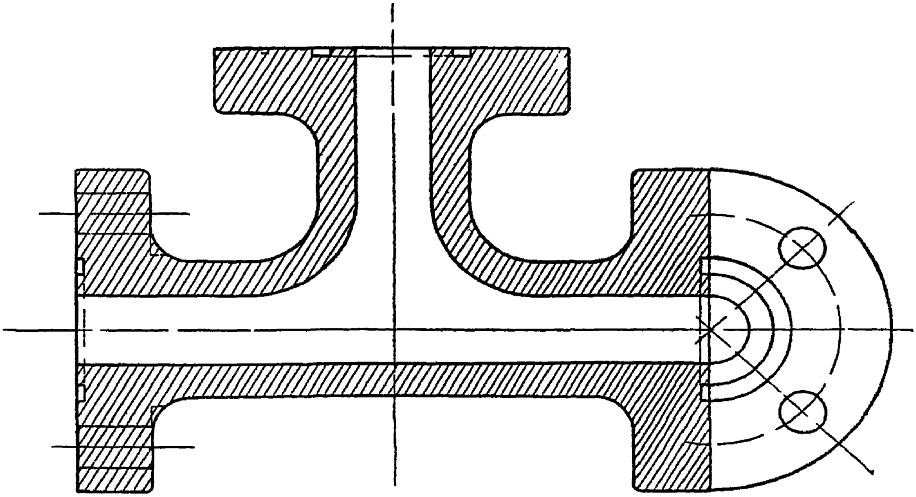

FIG 239

valve is such that air is admitted through a port cast in the side of the cylinder cover Any arrangement of piping is to be avoided which renders difficult the removal of a cylinder cover hence the provision of an elbow on the latter In other designs this elbow is cast integrally with the cover itself

With marine engines all cylinders are provided with starting valves, to which the air is led through a steel main pipe line running the whole length of the engine. Fig. 238 shews the type of pipe flange most commonly used, the material being steel. The tee pieces for distribution to the several cylinders may be of cast iron or cast steel. If the former material is used the design should be very substantial, as in Fig. 239. The pipe lines should be securely clipped to the framework of the engine, otherwise there is liable to be severe vibration, due to the surging of pressure within the pipe.

In large slow speed engines the diameter of the starting pipe may be about 0.07 to 0.1 of the cylinder diameter. In small high speed engines it is advisable to give the main distributing pipe a diameter of about 0.15 to 0.17 of the bore, in order to secure rapid acceleration.

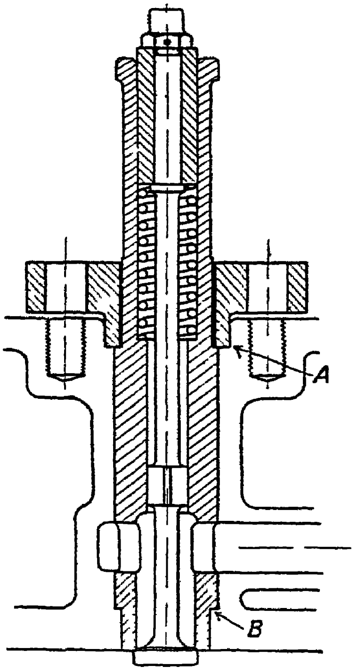

Fig. 240

Starting Valves.—These are usually located in the cylinder cover and operated by cams and levers in the same way as the other valves. An arrangement less frequently used consists of a centralised air distributing box of rotary or other type remote from the cylinder covers but connected to them by distributing pipes. Loss of compression is obviated by the provision of non return valves in the cylinder cover. The centralised distributing box may consist of a sleeve rotating in a casing in such a manner that a slot in the sleeve admits air successively to a number of ports communicating with the several cylinders. In other arrangements a set of cam operated mushroom valves is used. These schemes have not become common practice and will not be discussed here in further detail.

A common type of starting valve is shown in Fig. 240. No provision has been made here to prevent leakage past the spindle, and if the latter is a good ground fit in the casing the leakage should not be serious in amount. In large sizes of valve additional tightness may be secured to advantage by

the provision of a number of small Ramsbottom rings It is usual to make the diameter of the piston part of the spindle the same as the smaller diameter of the valve head The minimum spring compression should be equivalent to the maximum starting air pressure acting on an even area equal to that of the valve seat The valve casing should be sub stantially proportioned to prevent distortion and consequent leakage at the seat or binding of the spindle It will be noticed that with this design of valve joints have to be made at A and B simultaneously There is no practical difficulty about this Joint A is usually made with a copper or white metal ring A slight modification is sometimes made by the introduction of two cone joints as in Fig 241 This also works well Fig 242 shews a type of starting valve in which the air is led to the top of the valve casing instead of being introduced through a port in the cylinder cover

A useful type of starting valve devised by the Burmeister & Wain Company for Diesel Marine Engines is illustrated diagrammatically in Fig 243 With this design the valve becomes inoperative when the air pressure is removed and resumes working as soon as the pressure is restored This results in a great simplification of the manœuvring gear (see Chapter XIV) by the elimination of mechanism which in some other designs is provided for the purpose of throwing the starting valves out of gear when the fuel is turned on The desired result is achieved by attaching to the upper end of the valve spindle an air cylinder and piston kept in constant communication with the air supply by means of holes through the spindle In the absence of air pressure the spring A is sufficiently strong to keep the piston B at the bottom of the cylinder C thus removing the roller from the range of opera tion of the cam D When pressure air is turned on the piston is forced to the top of the cylinder and the valve remains operative so long as the force required to open the valve is less than the difference between the pressure load and the spring load on the piston B

Diameter of Starting Valves—On theoretical grounds the necessary diameter of starting valves would appear to depend on the pressure of the air supply amongst other things It so happens however that in those cases where a low pressure air system is the most convenient (viz in large marine instal lations) the multiplicity of cylinders to which starting air is

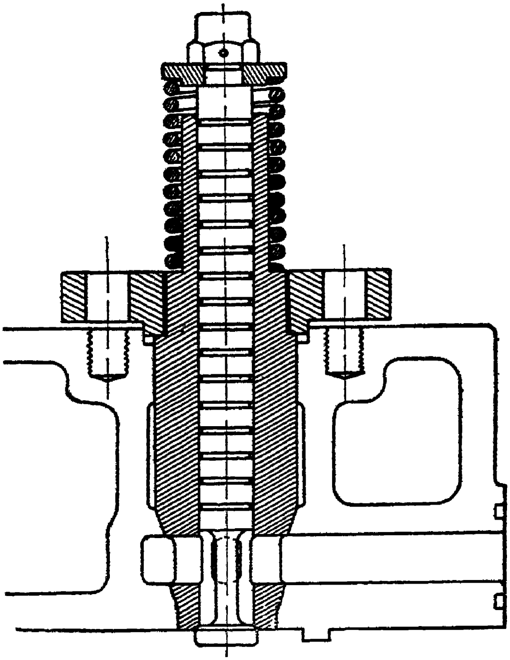

FIG 241

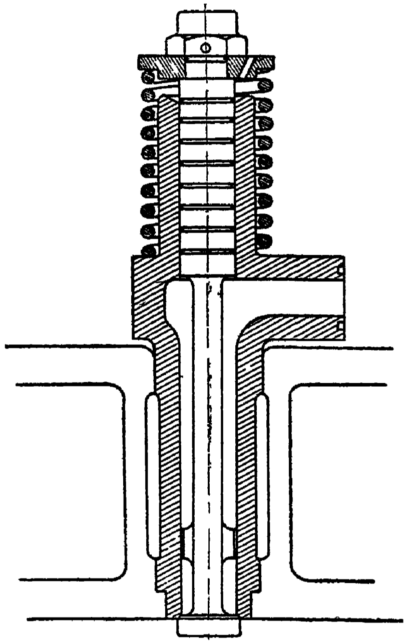

FIG 242

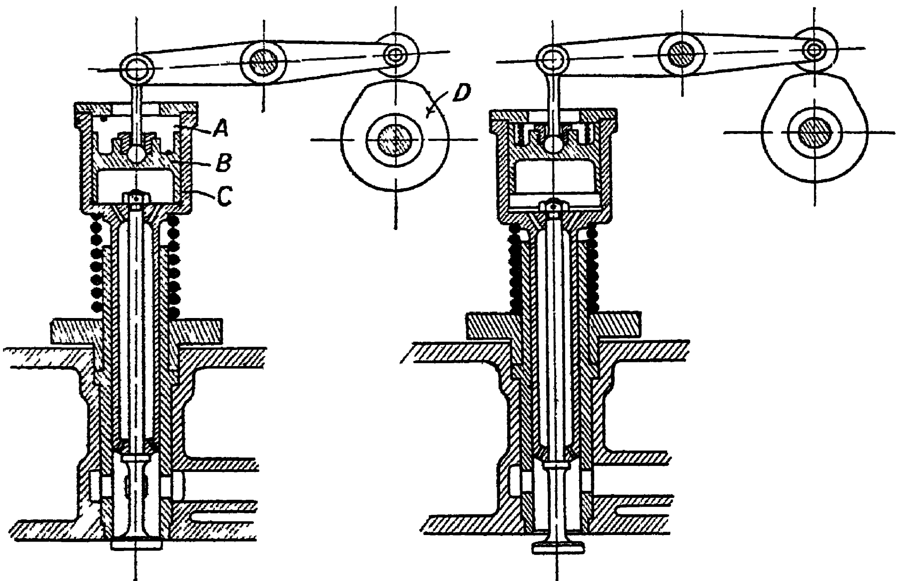

FIG 243

supplied affords adequate starting torque with a relatively low mean starting pressure in each cylinder The result is that roughly the same diameters of starting valve are used in either case i e whether a low or high pressure starting system be adopted typical figures being from about 0 1 of the bore in the case of large engines to 0 13 in small engines

With slow speed land engines it is very desirable to obtain a fat starting card to overcome the inertia of the heavy fly wheels which are usually necessary An engine in good working order should start firing in the first or second revolution on starting up cold It seems probable that in the event of low pressure air being used for such engines it might be necessary to fit starting valves to all the cylinders or to make the diameter of the latter larger than is customary with the high pressure starting air systems at present in use

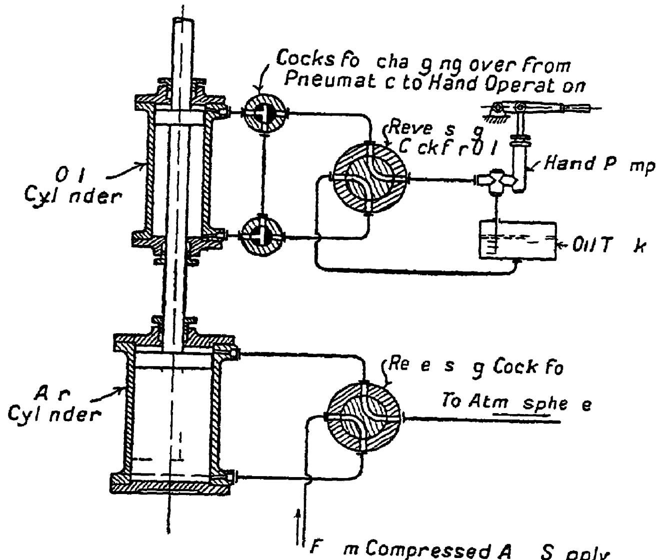

FIG 244

Air Motors —In large marine engines the work required to effect reversal of the valve mechanism when going from ahead to astern or vice versa is generally too great to be done with sufficient rapidity by a hand gear except in case of a break down of the air motor which is usually provided for the purpose In different designs the air motor takes various forms of which some are mentioned below —

(1) A small reciprocating engine (double acting) with two cylinders and cranks arranged at right angles The arrangement is almost exactly similar to the small auxiliary steam engines used on steamships for the reversing gear or the steering gear Low pressure air is used and the air motor is geared down by worm and worm wheel so that it makes a considerable number of revolutions for one movement of the reversing gear

(2) A single cylinder with piston and rod the reversing motion being performed in one stroke This arrangement is suitable for high or low pressure air and in either case the piston rod must be extended into an oil dashpot cylinder to reduce shock If the two ends of

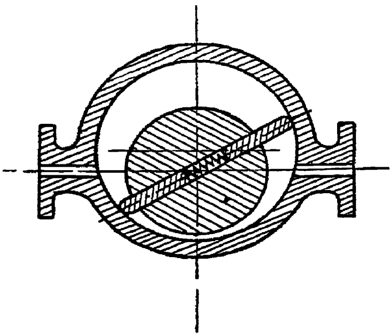

FIG 245

the oil cylinder be connected to a hand pump the latter may be used for reversing in the event of a failure of the air cylinder This scheme is illustrated diagrammatically in Fig 244

The reciprocating motion of the piston rod may be converted into rotary motion (one complete revolution or more) by a rack and pinion

(3) A rotary engine of the type which is frequently used as a pump in connection with machine tools motor cars and other small machines and which is shown diagrammatically in Fig 245 This type of motor is only suitable for low pressures and is arranged to do its work in a considerable number of revolutions by means of worm gearing

Types (1) and (3) would appear to have the advantage of greater adaptability to varying pressures It is an easy matter

to geaı the motor down so that ıt wıll turn under the lowest aır pressures antıcıpated At hıgher pressures wıre dıawıng at the ports pıevents the attaınment of an undesırably hıgh speed

In type (2) hydraulıc leather packıngs aıe used and consıder able care should be taken ın the desıgn and the workmanshıp to elımınate all unnecessary sources of frıctıon The strıct alıgnment of the two cylınders and the guıded end of the rod deserve specıal attentıon

Lıterature —Ford J M Hıgh Pressure Aır Compressors —Papeı read before the Greenock Assoc of Shıpbuılders and Engıneers See *Engıneerıng* October 20th 1916 *et seq*

CHAPTER XIV

VALVE GEAR

Cams —Wıth few exceptıons the valves are operated by external pıofile cams made of cast ıron chılled and ground on the face In order to facılıtate the removal of valves and cylınder covers ıt ıs usual to arrange the cam shaft to one sıde of the cylınders and to transmıt motıon from the cams to the valves through levers or a combınatıon of levers and push rods or lınks The arrangements ın general use gıve an approxımate one to one leverage between cam and valve and the figures

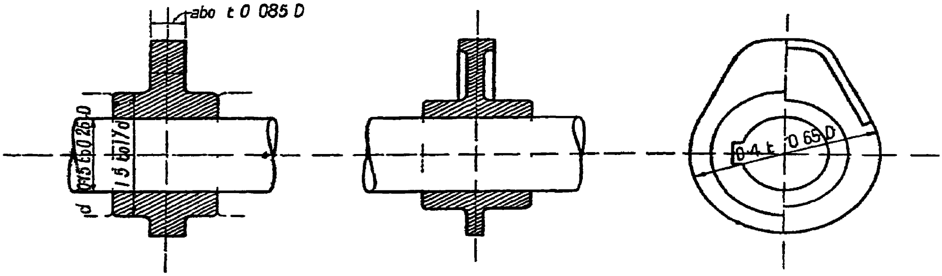

FIG 246

gıven above foı the wıdth of cam face are based on thıs pro portıon

Two forms of cam body for the aır and exhaust valves of four stroke non reversıble engınes are ıllustrated ın Fıg 246 the dımensıons beıng expressed ın terms of the cylınder bore Fıg 247 shews a combıned ahead and astern cam for a large marıne engıne The bosses should be bored a hard drıvıng fit on the cam shaft and theır lengths should be machıned accurately to dımensıons so that the complete group of cams requıred for one cylınder gıve correct spacıng when drıven hard up sıde by sıde

The fuel cam has to be of specıal constructıon on account of

the necessity for precise adjustment of the timing and a typical form is shewn in Fig 248 The toe piece is preferably made of hardened steel but chilled cast iron is sometimes used

Profile of Cams —The design of cam profiles for air exhaust and scavenge valves is a matter of reconciling the claims of the following desiderata —

(1) Rapid and sustained opening
(2) Absence of wear and noise

In slow speed engines the question of noise hardly arises and wear is easily kept to a reasonable minimum by adequate width of face For such engines the tangent cam shewn in Fig 249 is suitable For high speeds a smoother shape as shewn in Fig 250 is desirable and the relatively slow opening may be compensated by earlier timing Such profiles are easily drawn by deciding on some arbitrary smooth curve of roller lift and plotting corresponding positions of the roller with respect to the cam as in Fig 251 Some designers are in favour of a sinusoidal form of roller lift curve With these smooth profiles peripheral cam speeds of five feet or more per second can be used with quite sweet running On theoretical grounds the cam profile should be based on the roller clearance circle as in Fig 252 but it does not yet appear quite clear whether the procedure has the practical advantages claimed for it

The starting air cams are best given a sudden rise on the opening side to minimise wire drawing (Fig 253)

Combustion valve cam profiles are a study in themselves and the final decision rests with the test bed engineers The effective period is usually about 48 or 50 crank shaft degrees or 24 to 25 cam shaft degrees The cam piece should how ever give a range about 25% in excess of this after allowing for the normal roller clearance to allow for lost motion in the gear The tangent profile shewn in Fig 249 is usually found quite satisfactory but other shapes are used

In order to avoid noise in two stroke engines it appears necessary either —

(1) To make the cams of smaller diameter than those of a four stroke engine of the same size or
(2) To provide double faced cams mounted on a half speed cam shaft

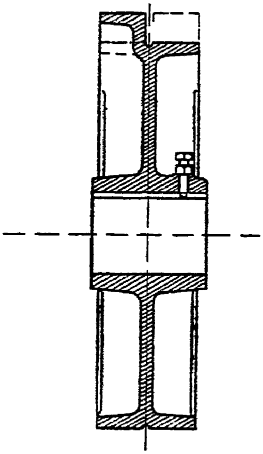

Fig 247

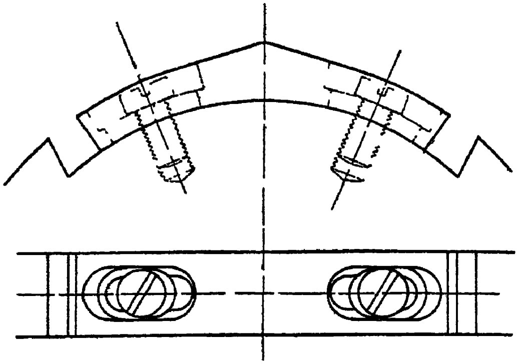

Fig 248

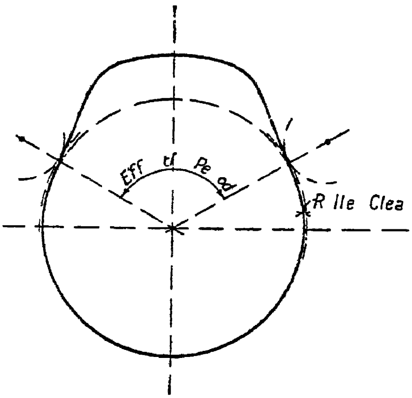

Fig 249

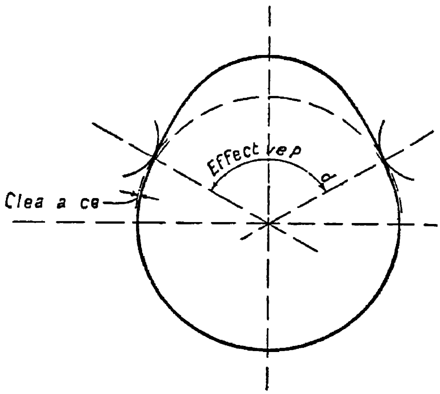

Fig 250

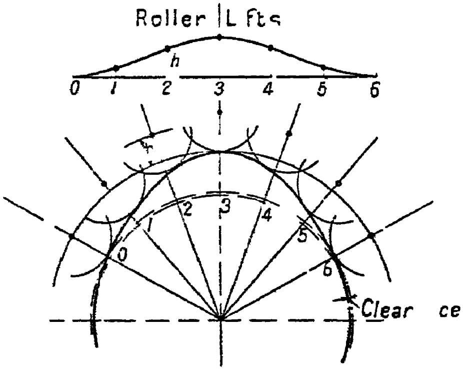

Fig 251

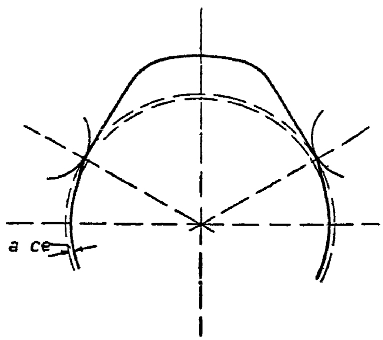

Fig 252

The first procedure is the more usual but the second would appear to have much in its favour In one marine design advantage is taken of this arrangement to work the engine on the four stroke cycle at slow speeds

Cam Rollers—Engines having longitudinally fixed cam shafts are usually provided with cam rollers of steel case

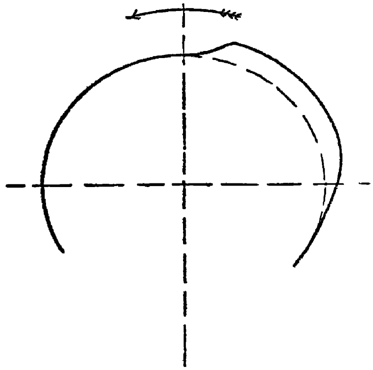

FIG 253

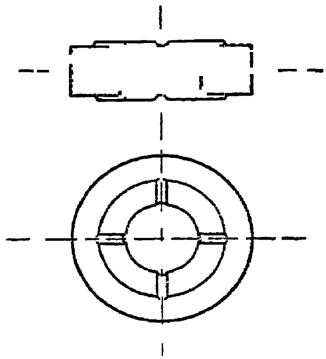

FIG 254

hardened and ground inside and out (Fig 254) and having a diameter of about one third that of the corresponding cams The grooves provided for hand lubrication of the pin should be noted In marine engines in which reversal of rotation is effected by the provision of ahead and astern cams mounted on a longitudinally movable shaft the rollers require to be of large diameter (about 60% of the cam diameter) in order that the idle cam may clear the lever as in Fig 255 Rollers of this size may be of cast iron bushed with phosphor bronze

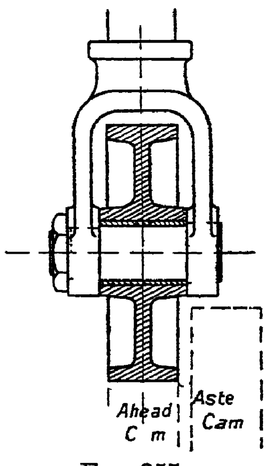

FIG 255

Valve Levers—A common arrangement of valve levers and lever fulcrum shaft for four stroke land engines is shewn in Fig 256 The fulcrum brackets are secured to the cylinder cover and the latter may be lifted complete with all valves and gear and replaced without disturbing the valve settings With the arrangement shewn it is necessary to lift away the fulcrum shaft and levers before the various valves can be removed for regrinding This very slight inconvenience

is sometimes overcome by means of split levers or by provision of horse shoe shaped distance collars on the fulcrum shaft which when removed leave sufficient space to allow the levers to be moved sideways clear of the valve casings These devices are desirable in the largest engines only Referring to Fig 256 below it will be noted that the fuel and starting levers are mounted on an eccentric bush A connected to the handle B The latter is provided with a spring catch engaging with notches in the fixed disc C in accordance with the following scheme and the diagram shewn in Fig 257

Top notch —Running —Fuel lever in its normal running position Starting valve roller out of range of cam

Middle notch —Neutral —Both fuel and starting valve rollers out of range of cams

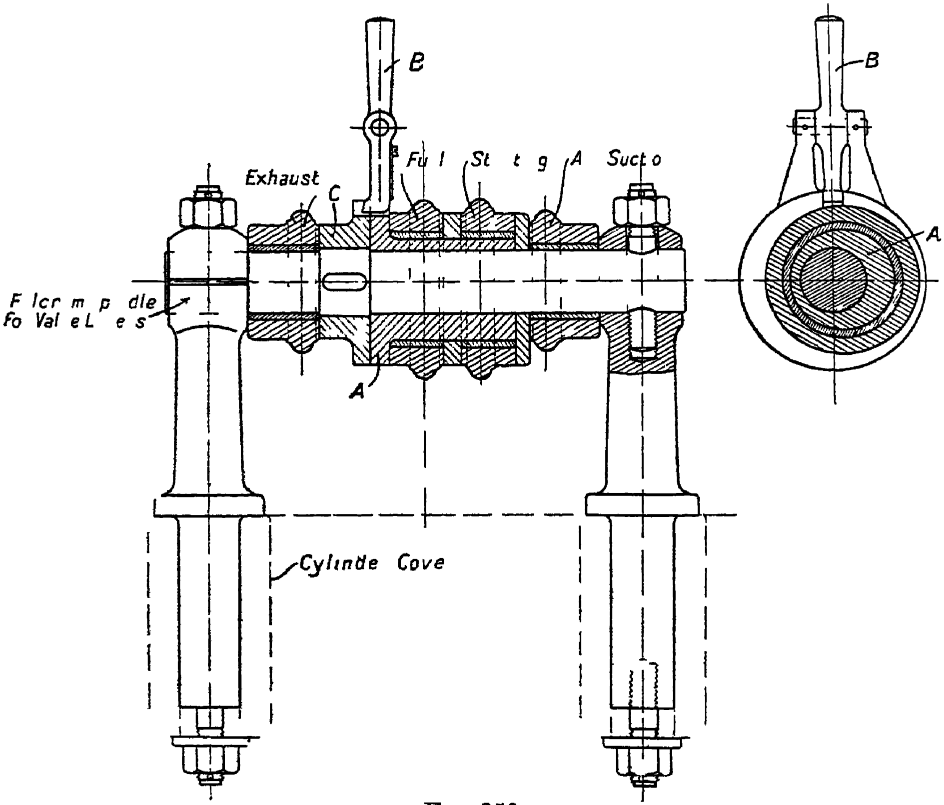

FIG 256

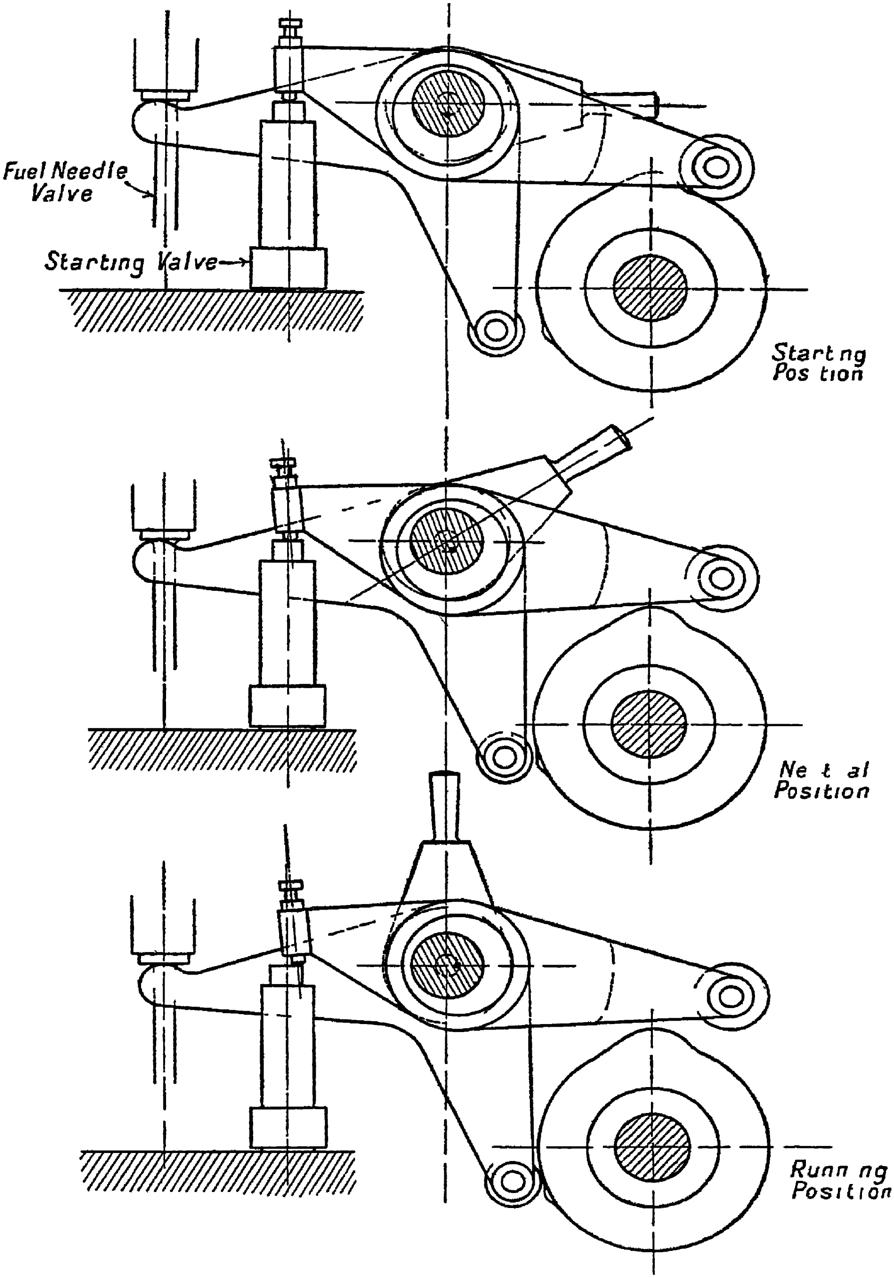
Fuel Needle Valve
Starting Valve
Start ng Pos tion
Ne t al Position
Runn ng Position

FIG 257

Bottom notch —Starting —Starting air valve lever in its working position Fuel valve roller out of range of cam

Sometimes this arrangement is modified by keying the eccentric bush and handle to the fulcrum shaft and allowing the latter to turn in its supports This scheme is useful when the disposition of the gear is such that an air suction or exhaust lever separates the fuel and starting levers This eccentric mounting of levers may also be used for exhaust lifting or to remove all the levers out of range of the cams during the axial displacement of the cam shaft of a reversing engine

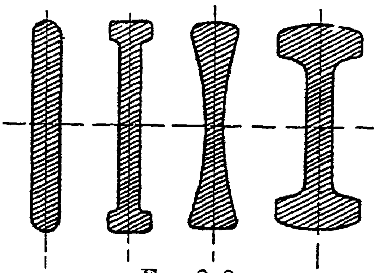

FIG 258

Certain well known marine makers of great repute do not consider it necessary to put the fuel valve out of action whilst the engine is running on compressed air and content themselves with suspending the supply of fuel to the valves during this period

The levers are generally of cast steel or malleable iron but good cast iron may be used if the stress is confined to about

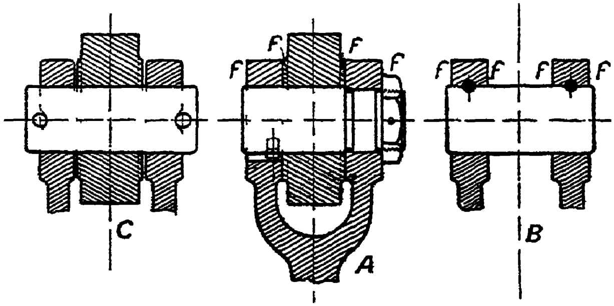

FIG 259

2000 lb /in ² Some alternative sections are shewn in Fig 258 The forked end of the lever calls for very little comment Type A (Fig 259) is a good design but expensive Type B is very commonly fitted and is open to little objection Type C is the cheapest existing construction and if accurate castings (machine moulded) are obtainable the only machining opera

tion required is to drill and reamer the hole for the roller pin The two grooves for the taper pin may be cast

In small engines the tappet end may consist of a plain boss screwed to receive a hardened tappet screw and lock nut as in Fig 260 In larger engines the more elaborate arrangements shewn in Fig 261 are usually adopted The bosses of the levers should be bushed with good phosphor bronze and provided with a dustproof oil cup of some description in order to reduce wear to a minimum With these precautions the bush should only require renewal at widely distant intervals

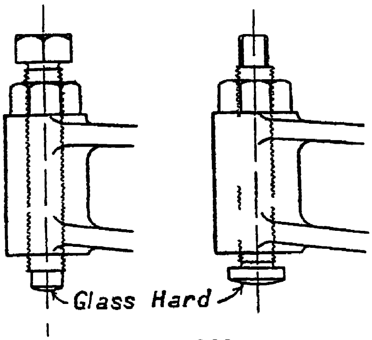

FIG 260

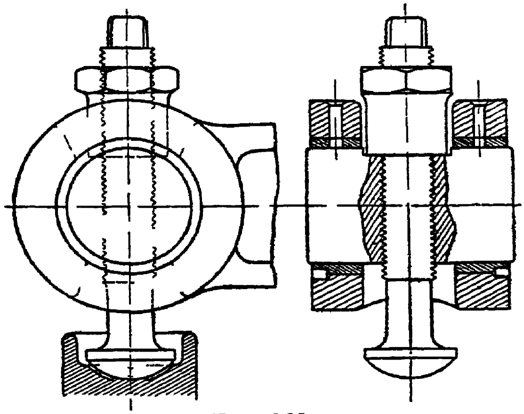

FIG 261

and means of adjustment are unnecessary even in the largest sizes of engines

Strength of Valve Levers—In four stroke engines the exhaust valve lever is the most heavily loaded Although the force required to operate the suction valve is relatively small it is usual to make the inlet valve lever of the same section as the exhaust lever for the sake of uniformity of appearance and the same pattern may frequently be used for both The loads imposed on the tappet ends of the various levers at the points of valve opening are given below —

FOUR STROKE ENGINES

Exhaust Valve About 45 lb per sq in of exhaust valve area + spring load + inertia of valve

Suction Valve Spring load + inertia of valve + a maximum of about 5 lb per sq in of valve area if the exhaust valve happens to be closing too early due to excessive roller clearance

Starting Valve 500 lb per sq in of valve area + spring load
Fuel Valve (a) Swedish type
1000 lb per sq in of needle area + spring load + inertia all reduced by the leverage employed
(b) Augsburg type
Difference between the spring load and 1000 lb per sq in of needle area at stuffing box

TWO STROKE ENGINES

Scavenge Valves Spring load less scavenge air pressure into area of valve + inertia of valve
Fuel and Starting Valves As for four stroke engines

All the above are of course subject to slight correction for friction

By way of example the main dimensions of the exhaust valve lever for a 20 four stroke cylinder are calculated below —

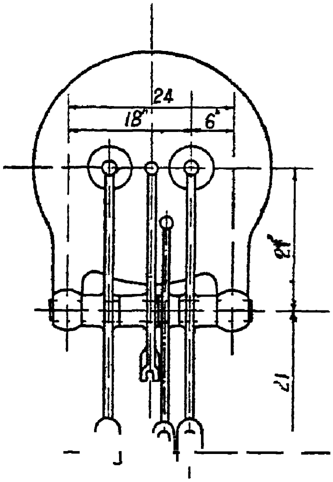

FIG 262

Data
Diameter of exhaust valve 6 5 ~~in~~ inches
Fulcrum spindle and exhaust valve lever centres as in Fig 262
Pressure load on exhaust valve
$=0\ 785 \times 6\ 5^2 \times 45 = 1490$ lb
Spring load at say 8 lb per in ² of valve area
$=0\ 785 \times 6\ 5^2 \times 8 = 265$ lb
Inertia load say 40 lb
Total load to open exhaust valve 1795 lb

Reaction at fulcrum spindle about 3600 lb

Bending moment at fulcrum spindle $= \dfrac{3600 \times 6 \times 18}{24}$ in lb

Allowing a stress of 6000 lb /in ²

$$\frac{d^3}{10} = \frac{3600 \times 6 \times 18}{6000 \times 24} = 2\ 7$$

$$d = 3 \text{ in}$$

Allowing for a bush $\frac{1}{4}$ thick and about $\frac{3}{4}$ metal at the boss the external diameter of the latter will be 5

Sketching in the approximate outline of the lever as in Fig 263 it is seen that at the weakest section AA the bending moment is about $1800 \times 18\ 5$ in lb and taking a stress of 5000 lb /in for cast steel the modulus Z of the section AA should be $\frac{1800 \times 18\ 5}{5000} = 6\ 66 \text{ in}^3$

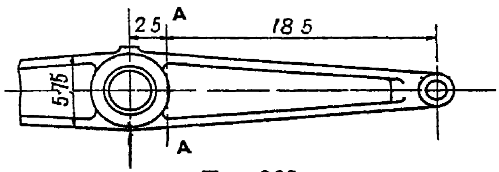

FIG 263

If the section AA is approximately T shaped as in Fig 264 then $Z = b\ t\ h$ nearly h is 5 75 and therefore

$$b\ t = \frac{6\ 66}{5\ 75} = 1\ 16 \text{ in}^2$$

which is satisfied by $b = 2\ 25$ and $t = 0\ 515$

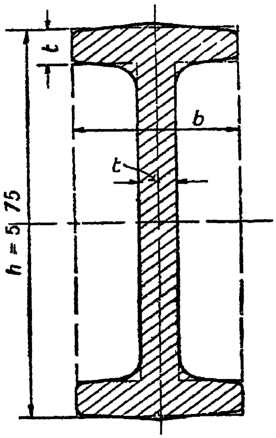

FIG 264

The lever may be made of approximately uniform strength by tapering towards the ends both in width and depth as in Fig 265 whilst the flange and web thicknesses are kept constant If a double bulb or other section is required for the sake of appearance and on casting considerations it is a simple matter to sketch in such a section approximately equivalent to the simple I section to which the calculation applies

Push rods —In some designs a push rod is introduced between the lever and the cam roller as shewn diagrammatically in Fig 220 *ante* in order to enable the cam shaft to be located at a low level For this purpose bright hollow shafting or even black lap welded steam tubes are suitable if not too highly stressed

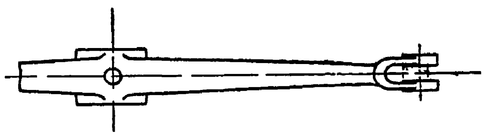
FIG 265

For handy reference in designing such push rods the following table taken from Prof Goodman s Mechanics Applied to Engineering is given for the buckling loads of tubular struts In using these figures it is advisable to use a factor of safety of

not less than about 3 or 4 and in no case to employ stresses exceeding 10 000 lb /in 2

BUCKLING STRESS (FREE ENDS) LB PER SQ IN

Ratio $\frac{\text{length}}{\text{diameter}}$	Mild Steel
10	59 000
20	42 000
30	29 000
40	20 000
50	14 000
60	10 500
70	8 200
80	6 500
90	5 500
100	4 500

The jointed ends of the rods may be of forged steel bar or malleable cast iron bushed with bronze as in Fig 266

Cam shafts —In modern shops the cam shaft may be rapidly and cheaply ground to size from black bars In order to facilitate the driving on of the cams for a multi cylinder engine the enlarged diameters are usually made of increasing sizes differing by successive thirty seconds of an inch or thereabouts as shewn exaggerated in Fig 267 The same figure which represents the cam shaft for a four stroke generating set of three cylinders also shews the method of supporting the shaft by means of a continuous trough with one bearing between each bank of cams This arrangement has a very neat appearance and makes pro vision for catching the oil which drips off the cams and rollers In some designs the cams are allowed to dip into an oil bath the level of which is maintained constant by a small pump

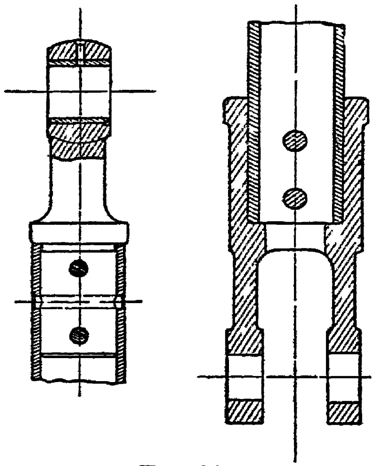

FIG 266

provided for the purpose or by a connection taken from the forced lubrication system A copious supply of oil to the cams has the advantage of securing quiet running

In other designs the cam shaft is supported by bearing brackets secured to the cylinders as in Fig 268 In this case

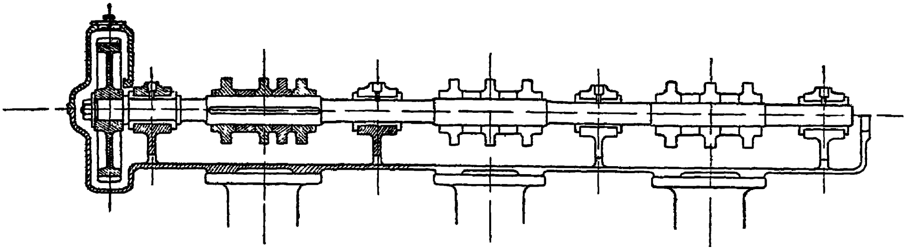

Fig 267

it is very desirable to fit light cast or sheet iron guards round each bank of cams The cam shaft bearings are divided horizontally for adjustment and the shells may be of cast iron lined with white metal solid gun metal or in small engines where the cost of material does not outweigh the advantage of simplicity of solid die cast white metal Owing to the slow peripheral speed and the intermittent character of the loading

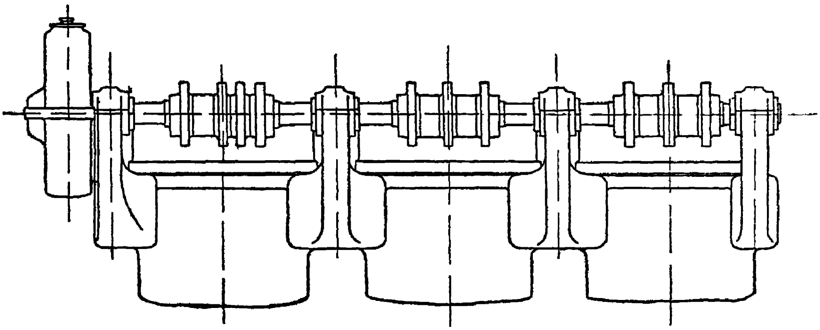

Fig 268

grease lubrication by Stauffer boxes is quite adequate although ring and syphon are more commonly used

A slightly different arrangement is shewn in Fig 269 Here the shaft is supported by a series of cam troughs one to each cylinder each trough having two bearings The extra rigidity of this arrangement allows of the cam shaft diameter being reduced below the figure required with the other arrange

ments described This division of the trough into segments is advantageous from the manufacturing point of view as the smaller parts are easier to cast and handle in the shops also one pattern serves for engines of any desired number of cylinders

Strength of Cam shaft —The size of cam shaft required for a given engine would appear to depend not so much on the stresses to which it will be subject as on the rigidity necessary to secure sweet running of the gear For a four stroke engine the opening of the exhaust valve against the terminal pressure in the cylinder is the severest duty which the cam shaft is

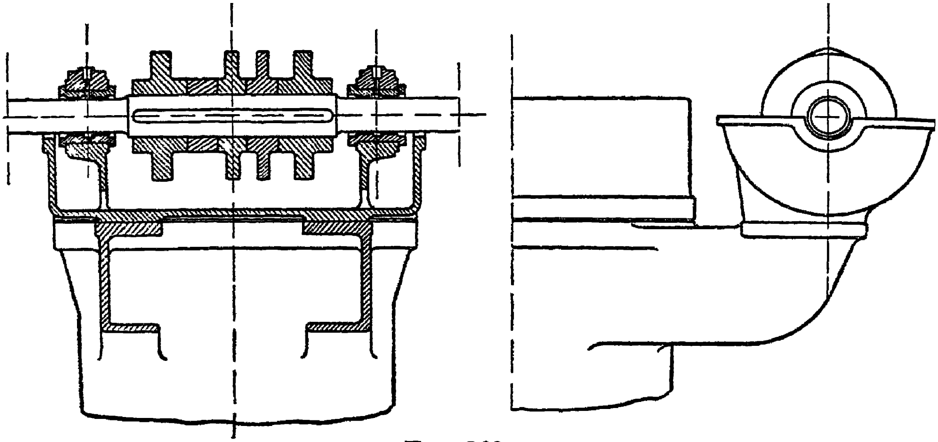

FIG 269

called upon to perform The load is applied and released fairly suddenly and a cam shaft lacking in torsional and trans verse rigidity would undoubtedly be subject to oscillations which in an acute case would give rise to the following evils —

(1) Noisy action of cams due to torsional recoil of shaft after each exhaust lift

(2) Interference with the timing of valves (particularly the fuel valves) of cylinders remote from the gearing end of the cam shaft

(3) Chattering of the gear wheels by which the shaft is driven

In view of the fact that as shaft diameters are increased the stiffness increases at a greater rate than the strength it seems just possible that strength considerations might outweigh those of stiffness in very large engines On the other hand

if angular deflection of the shaft between contiguous cylinders be accepted as the criterion then considerations of similitude give shafts of diameters bearing a constant ratio to the cylinder bores (or rather exhaust valve diameters) and constant stresses in all sizes if the terminal pressure is always the same The fact that in practice relatively thinner cam shafts are used in large engines may be due to the lower terminal pressures obtaining in the cylinders of the latter

The following table shews the approximate diameters of cam shafts used in practice on four stroke engines of different sizes —

Bore of Cylinder in inches	6	10	15	20	25	30
Diameter of Cam shaft in inches	1^{1}	$2\frac{1}{4}$	$2\frac{7}{8}$	$3\frac{3}{8}$	3 [illegible]/8	$4\frac{1}{4}$

The above figures hold for any number of cylinders up to four with the cam shaft drive at one end or eight with the cam shaft drive at the centre

For two stroke engines these diameters may be materially reduced on account of the absence of exhaust valves Average figures for existing practice appear to be about 25% lower than those given above for four stroke engines The fuel pumps and cylinder lubricating pumps are frequently driven off the cam shaft but any auxiliary gear such as circulating pumps etc requiring appreciable power are precluded

Cam shaft Drives —For non reversible engines the spiral drive shewn diagrammatically in Fig 270 is the favourite This drive comprises the following components —

(1) Lower spiral wheels
(2) Footstep bearing for vertical shaft
(3) Vertical shaft and couplings
(4) Upper spiral wheels

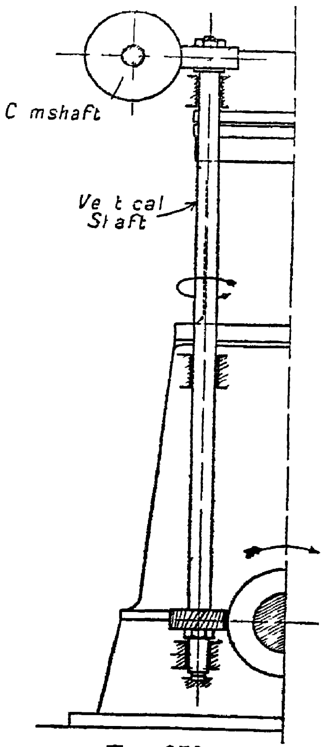

FIG 270

The lower spiral wheels generally have a 1 1 ratio so that the vertical shaft runs at engine speed In some designs the ratio is $1\frac{1}{2}$ 1 and the vertical shaft runs

at 50% above engine speed With the former arrangement the upper wheels have a ratio of 1 2 and with the latter 1 3

The construction of the footstep bearing has already been commented on in Chapter VII The vertical shaft is usually made the same diameter as the cam shaft or a trifle less and for convenience in dismantling is sometimes made in two or three pieces connected by couplings examples of which are shewn in Fig 271 The spiral drive has also been used for reversible two stroke engines as it lends itself to a particular type of reversing motion to be described later

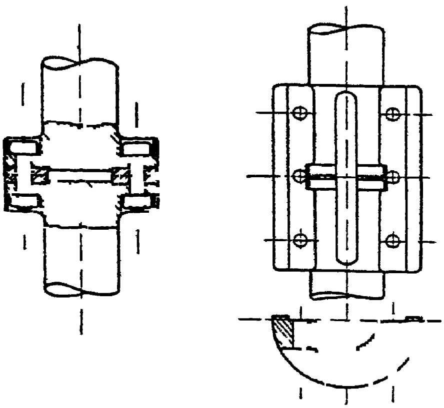

FIG 271

A combination of spiral and bevel drive is also used as shewn diagrammatically in Fig 272 and leads to a compact arrangement of valve gear

Reversible marine four stroke engines are usually fitted with some form of spur wheel drive in order to enable the cam shaft to be moved longitudinally without undue complication A floating cam shaft with spiral drive necessitates the use of a splined seating for the upper spiral wheel and the durability of such devices on a large scale seems questionable apart from the question of cost

The drive illustrated in Fig 273 shews a simple train of spur wheels and that of Fig 274 two pairs of spur gears connected by coupling rods the latter being driven by cranks at right angles This gear works very sweetly and appears to give perfect satisfaction in service

Combinations of spur and bevel gears have sometimes been used but would appear to have little to recommend them

The various gear wheel casings deserve careful attention in design and some of the points to be considered are enumerated below —

(1) The bearing or bearings incorporated with the gear case to be adequately connected to the framework of the engine and well lubricated

(2) Provision to be made for taking the end thrust of the

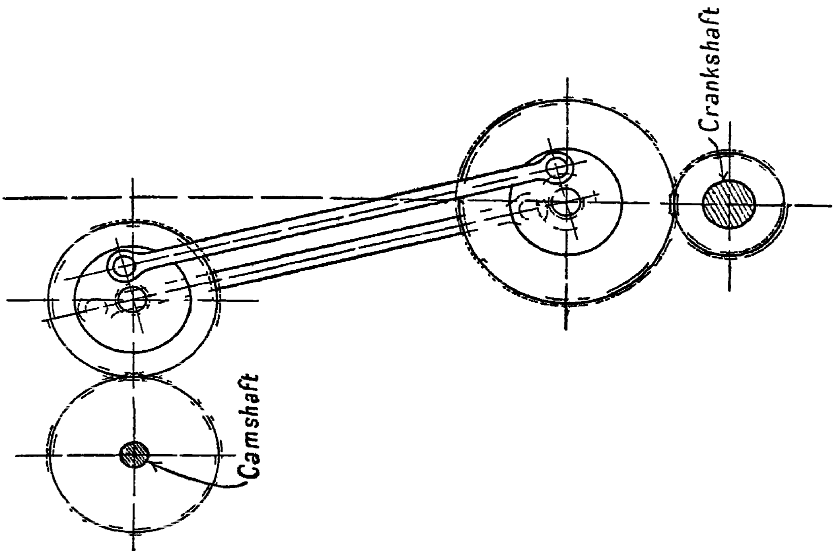

Fig 274

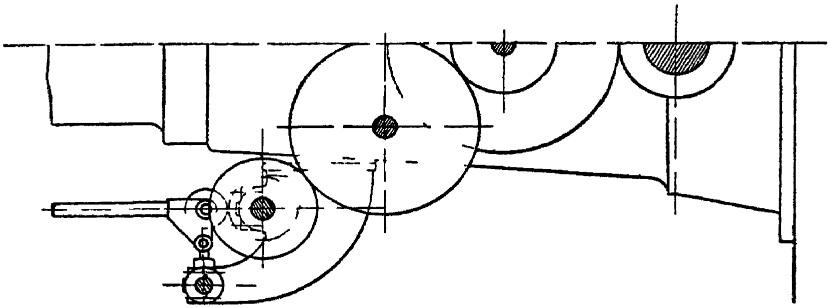

Fig 73

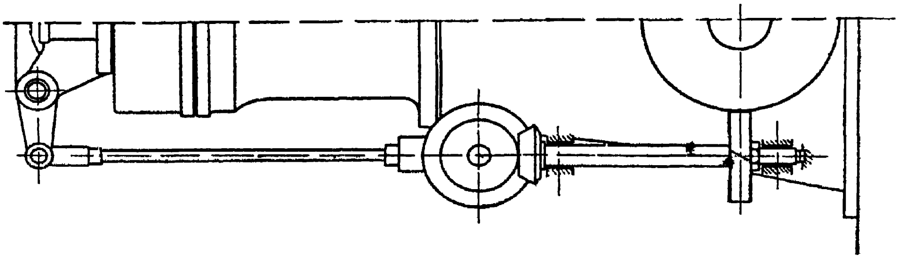

Fig 272

cam shaft due to spiral wheels etc preferably by ball thrust washers

(3) The wheels to run in a bath of oil and suitable arrangements to be made to prevent leakage of the latter

(4) The general arrangement of gear box and bearings to be compact and in general conformity with the design of the rest of the engine

Probably the simplest way of fulfilling the above requirements is to cast the gear case *en bloc* with a continuous cam trough as in Fig 267 *ante* or in the case of a central drive to suspend the gear case between two sectional troughs as in Fig 275 by a sufficient number of fitted bolts

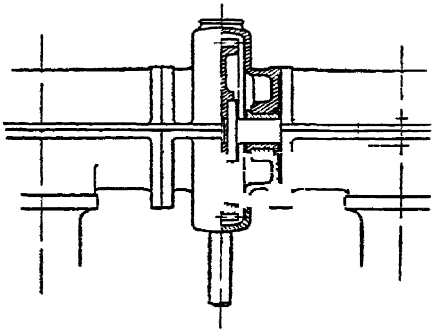

FIG 275

Spur and Spiral Gear for Cam shaft Drives —The question of the strength of the teeth hardly arises in this case and the problem consists in the selection of materials and proportions giving quiet running and absence of wear The following pairs of materials are in common use —

	Driver	*Follower*
(1)	Cast iron	Cast iron
(2)	Steel	Cast iron
(3)	Steel	Bronze

Of these the pairs (1) and (3) appear to give the best results with proper proportions and adequate lubrication etc

In good practice the normal circular pitch of the teeth is made about equal to one twelfth of the cylinder bore for both spur and spiral gears and the width of face about one fifth of the cylinder bore in the case of four stroke engines It is a fairly safe rule to make the pitch as coarse as the smallest wheel will allow in the case of spiral wheels With spur wheels fine pitches are not so objectionable as the sliding between the teeth is much less

For satisfactory running the teeth must of course be properly cut and the wheels accurately centred The tooth clearance should not exceed about 2/1000 and should be uniform all round

For particulars of tooth gearing calculations the reader is referred to the special books devoted to this subject The diagram shewn in Fig 276 is very useful in the preliminary stages of spiral drive calculation Suppose it is desired to design a pair of right angle spiral wheels to say 1 2 ratio first calculate the diameters of a pair of spur gears of the desired pitch and giving the desired ratio viz 1 2 Draw OA and OB equal to the pitch radii of the follower and driver respectively Complete the rectangle OBCA and draw any line DCE cutting the axes in D and E Then DC and CE will be equal to the pitch radii of equivalent spiral wheels having

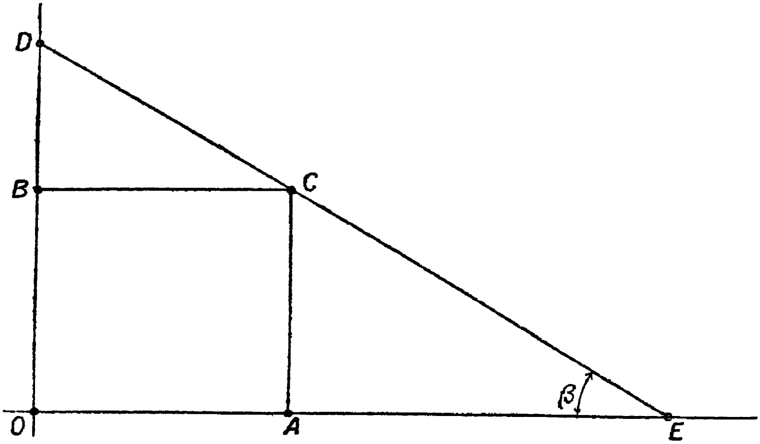

FIG 276

spiral angles α and β and having a normal pitch the same as the circular pitch of the spur wheels first calculated It usually happens that the wheel centres (DE) are fixed within approximate limits by space considerations and a process of trial and error is required to find suitable values for the number of teeth and the spiral angles The latter should not be less than about 27 as the efficiency falls off rapidly as this figure is reduced Having obtained an approximate solution by the above method the angles should be determined to the nearest minute by logarithmic trial and error calculation by means of the following relation —

$$\frac{AC}{\sin\beta}+\frac{BC}{\cos\beta}=DE = \text{required wheel centres}$$

Reversing Gears —In spite of early anticipations of diffi culty reversing gears for Marine Diesel Engines have attained a high degree of efficiency On the score of simplicity relia bility and quick action they compare favourably with the corresponding parts of steam engines A very great number of different gears have been suggested and patented but those in widespread use fall into two or three well defined classes which will be described below

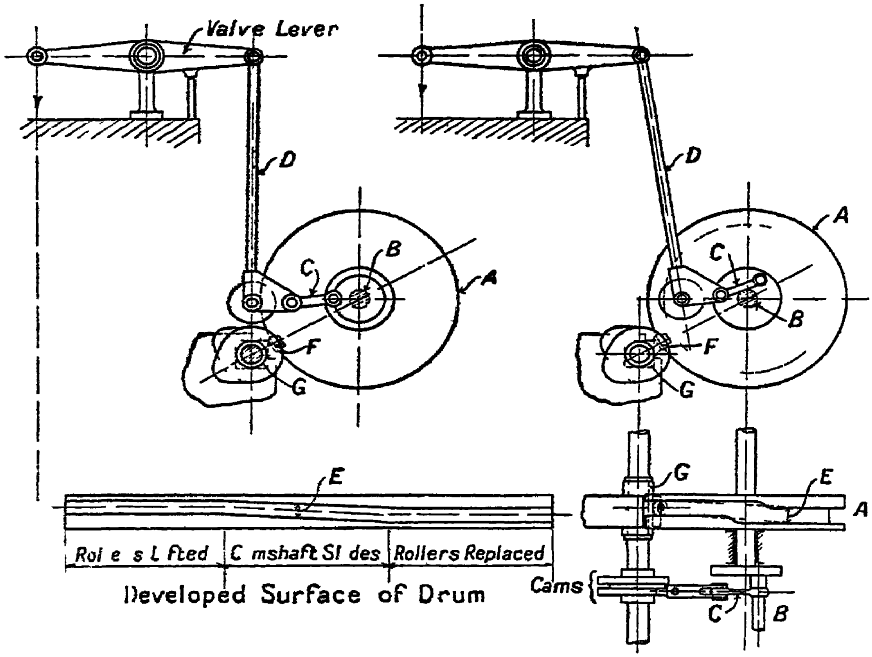

FIG 247

Sliding Cam shaft Type of Reversing Gear —This type of gear is the favourite for four stroke engines though it is equally applicable to those working on the two stroke cycle Ahead and astern cams side by side are provided for the opera tion of each valve Reversal is effected by sliding the cam shaft a few inches endways in its bearings so that the ahead cam is removed from the action of the roller and replaced by the astern cam and vice versa It is in general necessary to arrange means whereby the valve rollers may be swung clear of the cam noses during the longitudinal movement of the shaft otherwise fouls would occur In some very small

engines the necessity for such provision is obviated by employ ing curved faced rollers adapted to slide up and down inclined faces between the ahead and astern cams respectively

The method adopted in some of the Burmeister & Wain engines is shewn in Fig 277 A drum A is mounted on a cranked shaft B on which are hinged drag links C connected to the roller end of the valve push rods D Drum A is provided with a groove E the developed shape of which is shewn in the figure This groove accommodates a roller F attached to a movable collar bearing G Shaft B is rotated in the direc tion desired (ahead to astern or astern to ahead) by

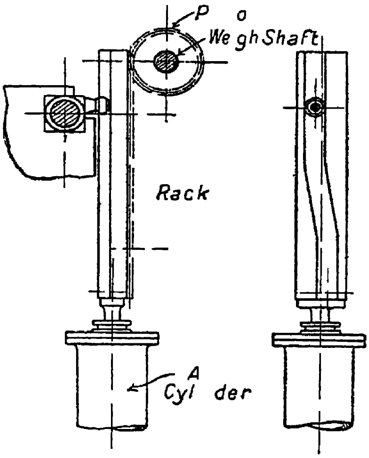

FIG 278

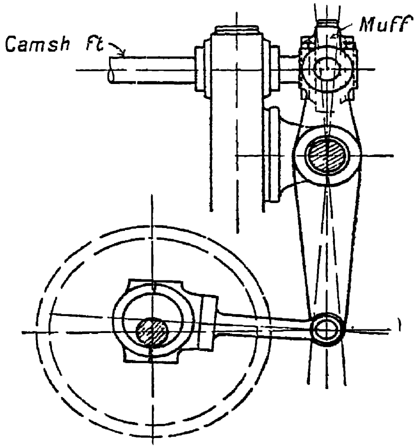

FIG 279

suitable gearing in connection with a reversing servo motor or the like Approximately one third of a revolution of the shaft suffices to swing the rollers clear of the cams meanwhile the cam shaft is stationary Another approximate one third of a revolution causes the groove E to shift the cam shaft from ahead to astern positions or vice versa whilst the valve rollers execute a harmless movement a little further out and back again The remainder of the revolution of the weigh shaft B replaces the rollers in their running position

In other engines of the same make the developed shape of the groove E is executed on the back of a rack by means of which the straight line motion of a vertical servo motor is converted into rotary motion of the weigh shaft This varia tion is shewn in Fig 278

If separate means be adopted for removing and replacing the rollers it is obviously possible to devise very simple means of shifting the cam shaft as in Fig 279 for example In such cases the two mechanisms must be interlocked to prevent a false manœuvre

Experiments show that the force required to move the cam shaft longitudinally is about one third of the weight of the cam shaft plus cams and other gear keyed thereto and this figure may be used as a basis of calculation for this type of gear It is advisable however to allow a fair margin of power as the resistance to motion must always be a matter of some uncertainty When the axial motion of the shaft has the

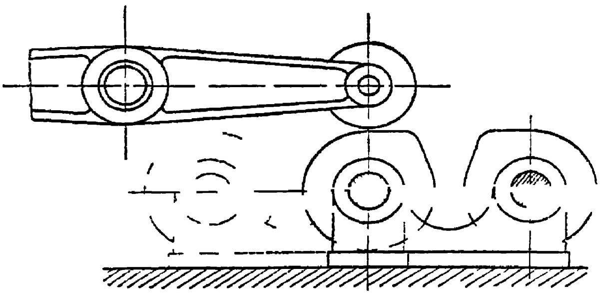

FIG 280

effect of opening one or more of the valves the resistance due to this cause must be added to that of the shaft itself

Twin Cam shaft Type of Reversing Gear —With this type of gear which is a speciality of the Werkspoor Company not only are separate cams provided for ahead and astern running but the latter are mounted on separate cam shafts capable of being slid into and out of action as required Fig 280 illustrates the arrangement diagrammatically The cam shaft drive is usually by means of coupling rods The chief advantage of this gear would appear to be the absence of special gear for swinging the rollers out of operation this process being unnecessary In recent Werkspoor engines this gear has been superseded by an arrangement of oblique eccentric bushes whereby the rollers are moved from the ahead to the astern cams and vice versa

Twin Roller Type of Reversing Gear —This gear depends on some form of link work such as that shewn in Fig 281 Rollers A and B lie in the planes of the ahead and astern cams respectively In the position shewn the timing of the valve

is controlled by the ahead cam and roller A roller B is meanwhile outside the radius of action of its cam Rotation of the weigh shaft C through a predetermined angle throws roller A out of action and brings roller B into action with the astern cam This type of gear has been applied to both four and two stroke engines

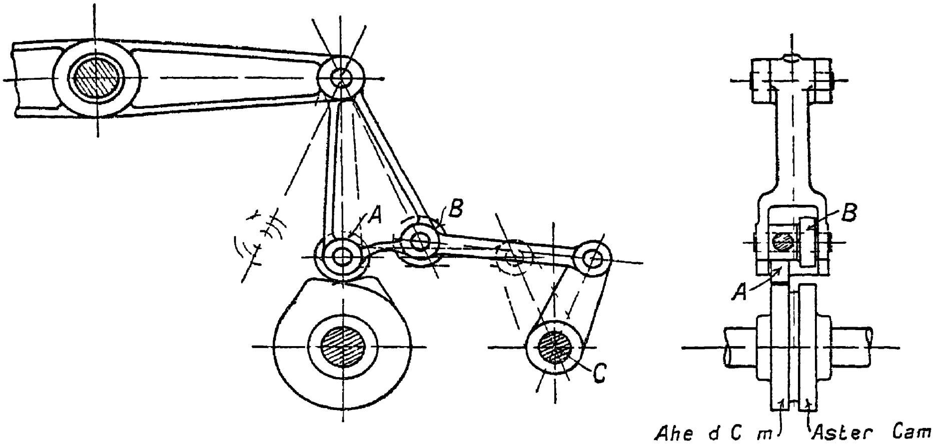

FIG 281

A different arrangement having some slight resemblance to the above is shewn in Fig 282 In this case there is only one roller which is swung from the ahead to the astern cam by a motion in a plane at right angles to the plane of the gear The roller face is curved to allow of this slight angular displacement from the vertical The inherent defects of this mechanism probably render it unsuitable for use in conjunction with any but the air starting valves

Selective Wedge Type of Reversing Gear —This ingenious gear illustrated diagrammatically in Fig 283 has been devised by Carels Freres and used in connection with the starting air and fuel valves of two stroke marine engines designed by them Ahead and astern cams are provided side by side and the valve roller A is wide enough to cover both Between the cams and the lever is interposed a roller wedge piece B under control of a cam C mounted on a manœuvring shaft D The latter is capable of independent rotary and endway motion A suitable rotary motion of the shaft D withdraws the wedge B to an extent which renders inoperative the ahead cam on which it rests A longitudinal movement of shaft D carries the

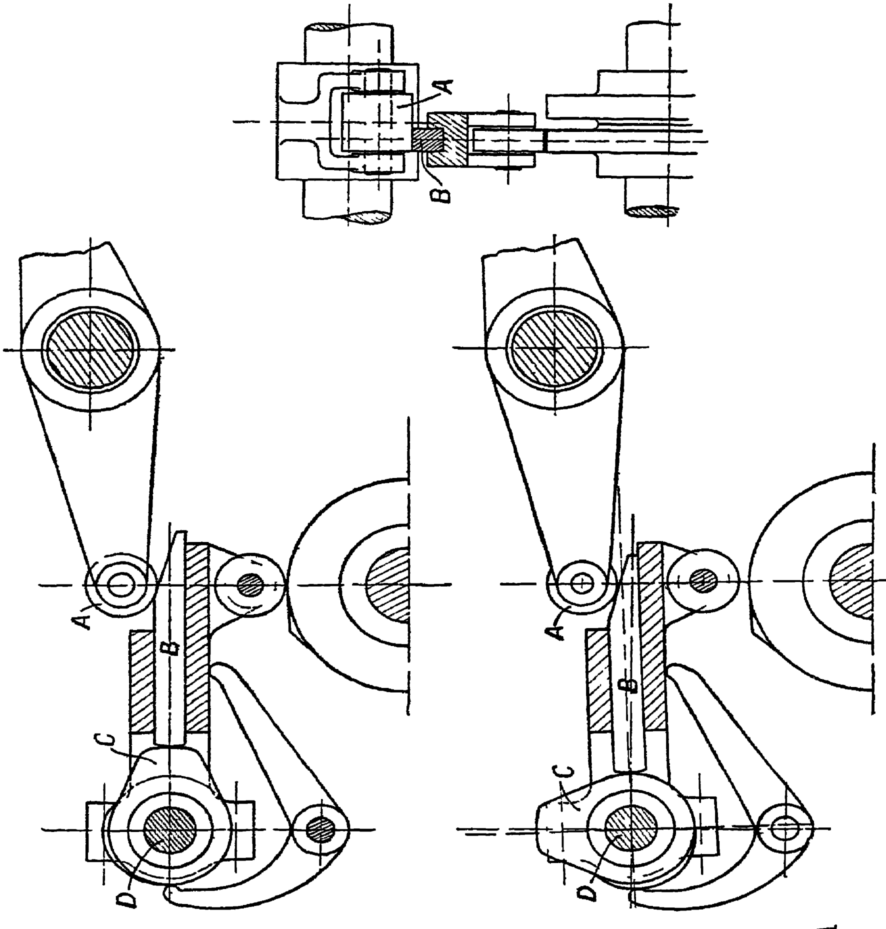

FIG 283

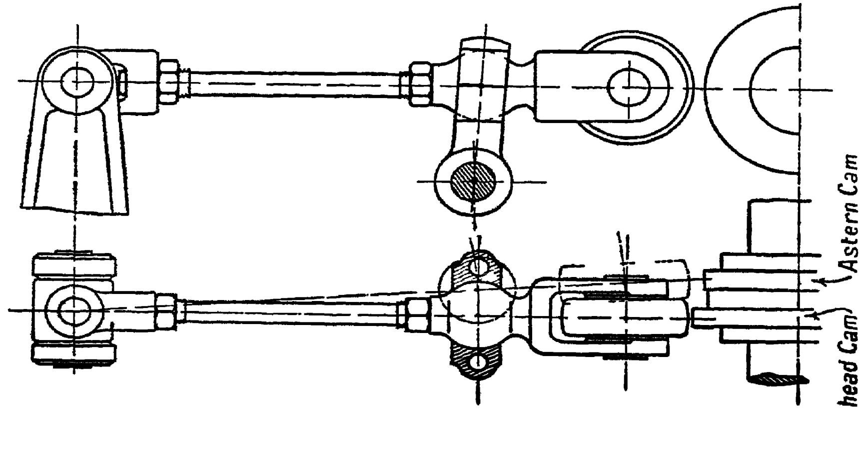

FIG 282

wedge B with it and a further rotation of D introduces the wedge between roller A and the astern cam and vice versa for astern to ahead It is to be noticed that by a suitable arrangement of the durations and sequences of the cams by which the wedges are operated the engine is caused to start up in any predetermined manner as for example —

Position (1)	Six cylinders on air	Fuel valves inoperative
(2)	Three cylinders on air	Three cylinders on fuel
(3)	Six cylinders on fuel	Air valves inoperative

Special Reversing Gear for Two Stroke Engines —With two stroke engines there are a number of means by which the duplication of cams may be avoided Considering the case of

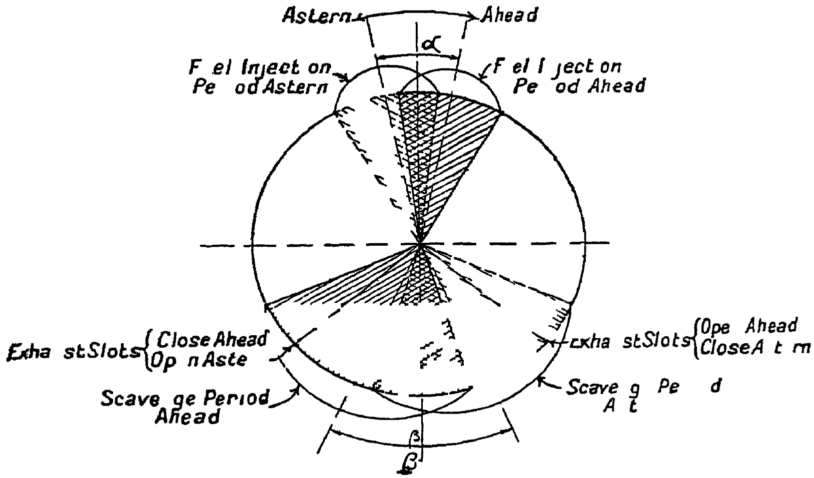

FIG 284

an engine fitted with scavenge valves and neglecting the starting air valves for the moment the valve settings for ahead and astern will be somewhat as shewn in Fig 284 It will be seen that for both fuel and scavenge valves all that is required to effect reversal is the rotation of the cam shaft through a certain angle α for the fuel valve and β for the scavenge valves In some early engines it was decided to select $\alpha = \beta =$ about 30 to 35 and so effect reversal of both valves by one movement In later engines however it is more usual to use the rotation of the cam shaft to reverse the scavenge valve only and adopt independent means such as duplicate cams etc for the fuel and starting valves The effect of the rotation of the cam

shaft on the settings of the latter must of course be allowed for in fixing the angular positions of the fuel and starting cams

A simple method of effecting the desired rotation of the cam shaft of a small engine is shewn diagrammatically in Fig 285 Spiral drives are used and the vertical shaft is in two pieces C and D connected by a splined coupling permitting vertical movement of the upper half D The vertical

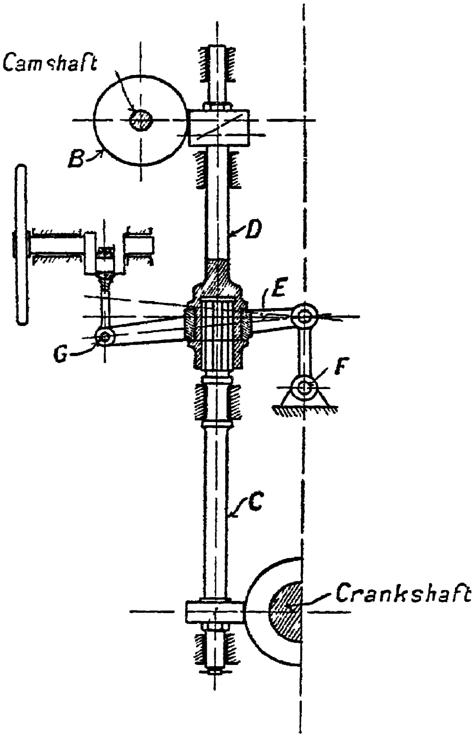

FIG 285

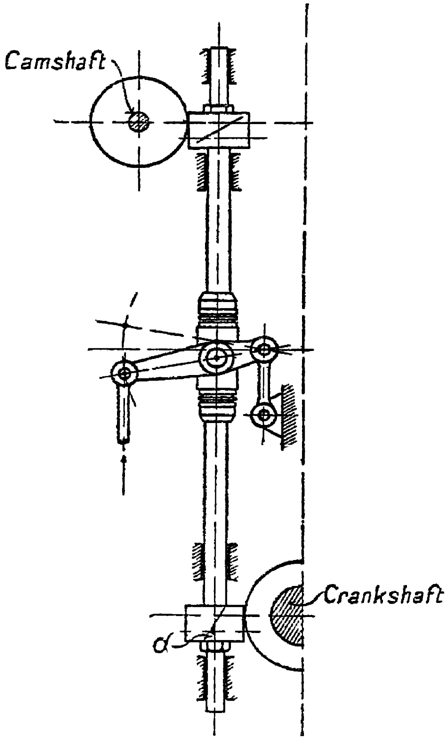

FIG 286

position of D is determined by the lever E which is hinged at F and connected at G to an eccentric or other suitable means of transmitting motion from the hand wheel The extreme upper and lower positions of D determine the ahead and astern running positions

If h =Lift of vertical shaft

s =Pitch radius of wheel B

Then $\frac{h}{r}$ =Reversing angle in radians

In other arrangements the vertical shaft is moved as a whole as in Fig 286 and in calculating the amount of motion required for a given reversing angle it is necessary to take into account the rotation of the vertical shaft due to the sliding between the lower helical wheels

Consider the case where both upper and lower gears have a ratio of 1 1 Let α be the spiral angle of the crank shaft gear wheel

Note that $\alpha < 45$

Let h = Lift of vertical shaft

r_1 = Pitch radius of vertical shaft lower wheel

r_2 = Pitch radius of cam shaft wheel

Then

Rotation of cam shaft due to axial movement of vertical shaft $= \frac{h}{r_2}$ radians as before

Further

Rotation of vertical shaft due to sliding of lower spiral wheels $= \frac{h \tan \alpha}{r_1}$

Therefore

$$\text{Reversing angle} = \frac{h}{r_2} \pm \frac{h \tan \alpha}{r_1}$$

With the arrangement shewn the positive sign applies when the upper and lower spirals have the same hand and the negative sign when they are of opposite hand

An inspection of the valve settings for ahead and astern as shewn in Fig 284 *ante* reveals the fact that the reversing angle is always described in the direction opposite to the previous direction of motion Advantage has been taken of this fact to obtain self reversing valve settings by arranging between the cam shaft drive and the cam shaft proper a claw clutch having angular clearance between the jaws equal to the reversing angle With this arrangement independent reversible gearing must be used for the starting air valves A suggested improvement on the above is to provide mechanical means for taking up the slack between the jaws whilst the engine is standing instead of allowing it to be suddenly taken up on starting

Movable Roller Type of Reversing Gears —Instead of rotating the cam shaft through a certain angle relative to the

cam roller the same effect may be obtained by turning the roller relative to the cam shaft

One such arrangement is shewn in Fig 287 It will be noticed that the valve lift is less in the astern position than in the ahead but this is unimportant A similar device is shewn in Fig 288 In both these designs the reversing angle is conveniently halved by fitting double nosed cams to a half speed cam shaft

Another gear coming under this category is shewn in Fig 289

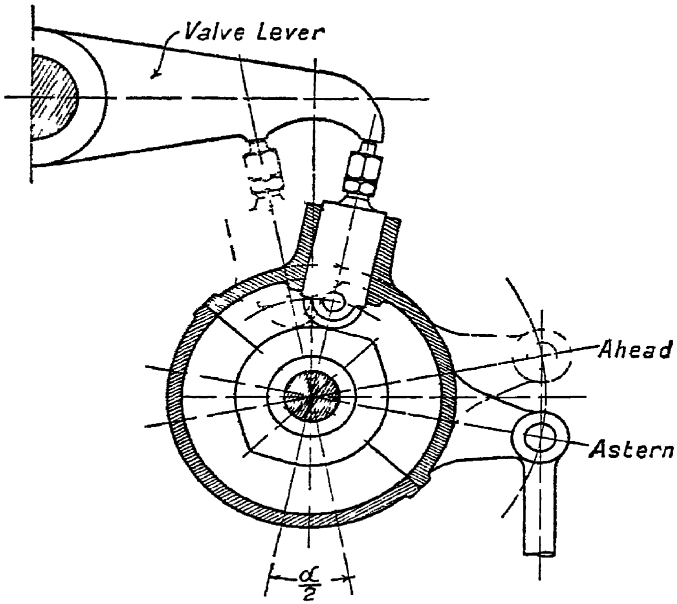

Fig 289

The displacement of the roller from its ahead to astern position is effected by the partial rotation of the eccentric fulcrum A and the roller passes through a neutral position in which it is outside the radius of operation of the cam

The above descriptions by no means exhaust the list of existing Diesel Engine reverse gears and doubtless others remain to be invented It is evident therefore that the problem of reversibility no longer presents any obstacle to the development of the Diesel Engine for marine service

For slow running engines there is probably little serious objection to any of the gears which have been described For

high speed engines however most existing gears for two stroke engines are noisy and of doubtful durability The sliding cam shaft scheme is equally applicable to two stroke as to four stroke engines and is perhaps as simple a solution as any

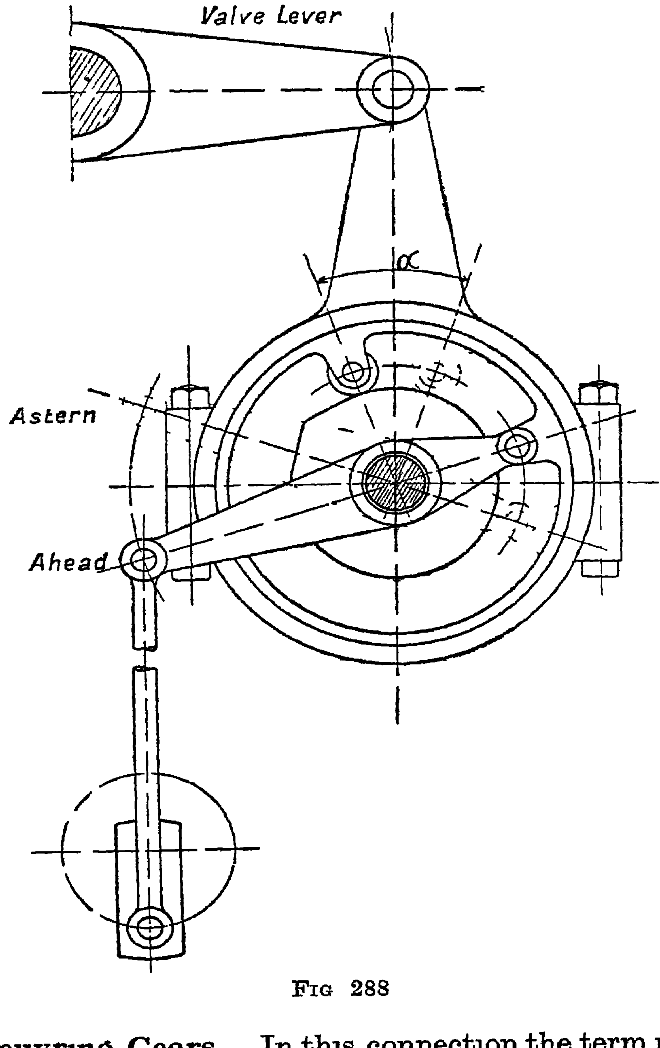

FIG 288

Manœuvring Gears —In this connection the term manœuvring gear is applied to those mechanisms apart from the reversing gear which come into operation on starting up a marine engine

The procedure differs in different designs but in general the following remarks are applicable —

(1) When the engine is standing the blast air supply should be cut off to prevent accumulation of pressure in any cylinder the fuel valve of which happens to be open If means be provided for putting the fuel valves out of operation in the stop position the blast cut out is not so essential but is still desirable as a safeguard

(2) The blast air should be turned on automatically immediately the engine is started although it is quite advantageous to provide an independent shut off and regulating valve under the control of the engineer

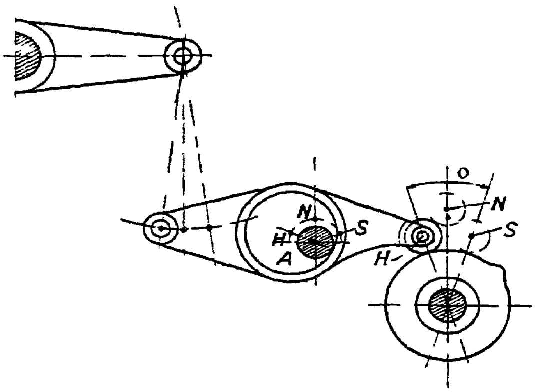

Fig 289

(3) When the engine is standing the starting air should be cut off as there is otherwise great loss of air due to leakage past the starting valves The starting air shut off may be automatic or hand operated if the latter it should be opened and closed by one simple motion An ordinary high pressure globe valve fitted with a quick threaded spindle is suitable for this duty

(4) The fuel pump suction valves should be held off their seats until such time as the fuel valves are in running position independent of the position of the fuel control

(5) The fuel control should be a handle (not a wheel) with a wide range of movement between no oil and full oil

(6) A wheel or better still a lever is provided in connection with suitable mechanism for putting the starting valves into

operation at starting and subsequently putting them out of operation when sufficient speed has been attained to ensure firing in the cylinders The same mechanism may or may not (in different designs) throw the fuel valve mechanism out of and into operation Furthermore in some designs the operation of this gear is graduated as in the following scheme which refers to a six cylinder engine —

First notch —Six cylinders on air (starting)
Second notch —Three cylinders on air Three cylinders on fuel
Third notch —Six cylinders on fuel

The gear under consideration is connected with the fuel pumps and with the blast starting air supply so that the following conditions are secured —

(a) Movement of the lever towards the starting position automatically turns on the starting and blast air
(b) Suction valves of all fuel pumps held off their seats

Further movement of the lever puts the starting valves of some or all of the cylinders out of operation and simultaneously allows normal operation of the corresponding fuel pumps and also of the fuel valves if these latter are arranged to be out of operation during the time the starting valves are working

(7) Some simple type of interlocking gear is usually fitted to prevent the following false manœuvres —

(a) Starting the engine before the reversing gear is in either the full ahead or full astern position
(b) Operating the reversing gear before the manœuvring gear has been put into the stop position

Some of the means adopted to secure the conditions outlined in sections (1) to (7) will now be described Further reference need not be made to the reversing gear as with the exception of the interlocking arrangements mentioned above the reversing arrangements are entirely independent of the manœuvring gear

A simple type of manœuvring gear is shewn diagrammatically in Fig 290 The fuel and starting levers are eccentrically mounted on fulcrum shafts as described earlier in this chapter Each fulcrum shaft A is connected by links and

levers to a manœuvring shaft B under the control of a hand lever C In the upper position of lever C all the fuel valves are in operation and in the lower position the starting air valves A link D connects the manœuvring lever to an eccentric fulcrum on the fuel pump by means of which the suction valves are lifted by suitable tappets provided for this purpose during such time as the starting air valves are in operation Another link performs a similar opera tion on the blast air control valve but in this case the connection is such that the blast air is only cut off in the neutral or stop position of the manœuvring lever

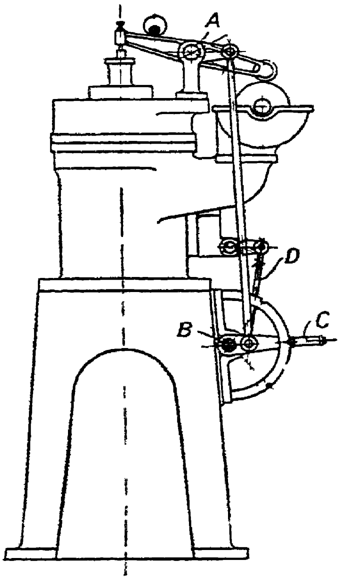

Fig °90

In some engines the above arrange ments are adopted in principle but two separate control gears and levers are provided for the forward and aft halves of the engine The two control levers are placed close together so that the engineer can work one with either hand On starting he pulls both towards him thus putting all cylinders under starting air As soon as sufficient speed has in his judgment been attained he pushes one lever towards the fuel position If firing starts he then pushes the other lever into the fuel position If on the other hand firing does not ensue he may pull back the lever into the starting air position and try the other lever in the fuel notch With an engine in good order it is probably advan tageous to throw over both levers simultaneously

In other designs it is not necessary to operate the fulcrum shafts as the starting valves automatically throw themselves into operation when starting air is turned on and become in operative when the starting air pressure is released

A great deal of ingenuity has been expended on the design of gears for throwing successive combinations of cylinders from air to fuel positions by a continuous movement of a wheel Some designers have even gone the length of combining the reversing and manœuvring mechanisms so that all positions ahead and astern are secured by clock wise and

anti clock wise rotation of this wheel In the writer s opinion such gears are not to be desired for the following reasons —

(1) Intelligent manipulation of machinery involves a certain parallelism between the mental state of the operator and the response which the machine makes to his control This state would appear to be most easily secured when separate and distinct operations on the part of the machine are made in response to separate and distinct movements on the part of the operator Hence it would appear best to keep the reversing and manœuvring control separate with the exception of what ever measure of interlocking is necessary to prevent accidents

(2) Gears of the kind referred to are usually complicated and not easily understood or overhauled

(3) The complication of such gears is not infrequently associated with backlash which renders accurate valve setting difficult to effect and maintain

(4) Complicated gears do not appear to have any practical advantages to offset their increased cost

Interlocking Gears —The precise form which an interlocking gear takes in any design depends on the forms of mechanism adopted for the reversing and manœuvring gears respectively but the problem very frequently reduces to that of two shafts either of which shall only be capable of movement in prescribed positions of the latter A simple interlock for two parallel shafts subject to partial rotation is shewn in Fig 291 It will be observed that the manœuvring shaft A can only be rotated when the reversing shaft B is in one of two positions (ahead and astern) defined by the positions of the gaps cut in the circumference of a disc keyed thereto Furthermore the shaft B can only be rotated from its ahead to its astern position (or vice versa) when the manœuvring shaft A is in one position—the stop position The solution when the shafts are at right angles as in Fig 292 is equally obvious An indefinite number of other schemes could easily be devised to meet the requirements of different arrangements of gear

Hand Controls —It is essential that all the wheels and levers by means of which the engine is controlled should be grouped together so that they may be manipulated by one man in one position In the best known designs there are a pair of long levers for controlling the groups of cylinders in

starting and passing over on to fuel In the Burmeister & Wain engines the same levers control the quantity of fuel delivered to the engine cylinders In some other designs a separate lever or wheel is provided for this purpose In almost every case separate wheels or levers are used for reversing The usual positions for the control station are at the centre of the engine on the bottom or top platforms

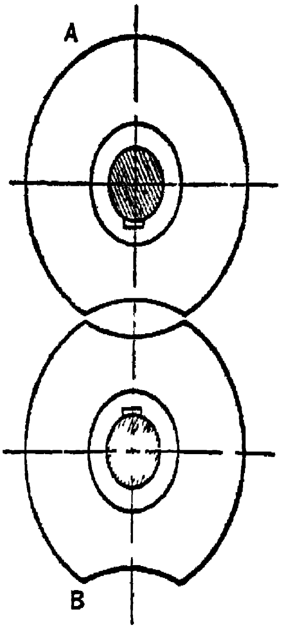

Fig 91

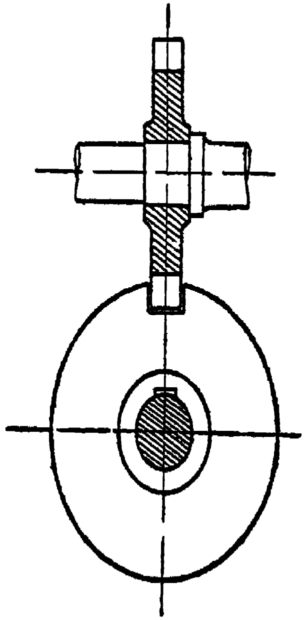

Fig 292

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